Marketing Research Report No. 921

A Study of Refrigeration Systems for Urban Food Distribution Centers

Agricultural Research Service
UNITED STATES DEPARTMENT OF AGRICULTURE

PREFACE

New concepts in food handling and distribution have brought food wholesalers together into large wholesale food distribution centers. Such centers have already been constructed in many cities, including Philadelphia, New York, Boston, San Francisco, and Atlanta, and others are anticipated in other cities.

Each new center faces the problem of selecting an optimum cooling system to handle the refrigeration requirements of all the individual firms. The need for information about cooling systems in order to select the best one for the situation prompted the Transportation and Facilities Research Division of the Agricultural Research Service to seek a study on "Conducting Investigations to Determine the Most Efficient and Least Costly Refrigeration System in Given Situations."

After careful consideration of several firms having experience in refrigeration system design for food storage facilities, contract No. 12-14-100-8311(52) was awarded to the York Division of Borg-Warner Corporation, York, Pa. The Borg-Warner Research Center, Des Plaines, Ill., and the Advance Engineering Group at York, Pa., provided personnel and facilities to conduct investigations and submit recommendations. The initial data and conclusions in this report were prepared by the York Division of Borg-Warner under this contract. Ralph McNatt, McNatt Engineering Company, and Austin Diehl, York Division, assisted the author in arranging and writing the final report.

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A Study of Refrigeration Systems for Urban Food Distribution Centers

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SUMMARY

A study was made to determine the most economical and efficient means of supplying refrigeration to a hypothetical, four-building food distribution center housing 34 firms in a wholesale food distribution center in Chicago, Ill. The firms buy and sell fresh fruits and vegetables, meats and meat products, poultry, eggs, and groceries. It was assumed that the 34 firms would need refrigeration for 2 rooms at −20° F., 11 rooms at −10°, 1 room at 25°, 17 rooms at 32°, 7 rooms at 40°, 1 room at 45°, 19 rooms at 50°, and 5 rooms at 72°.

The most economical choice was to have one central refrigeration system (Situation II) for the entire complex. Second choice, costing 8.2 percent more to install and 25.4 percent more to own and operate, was to use a central system in each of the four buildings (Situation III). Least desirable was to have each firm buy its own individual refrigeration system (Situation I). This would cost 4.3 percent more to install and 61.9 percent more to own and operate than the single central system, and it would not provide heating and air conditioning for offices.

The study considered the climate of the area, initial capital expenditures, thickness of insulation needed, type of refrigerant used, electric power costs locally, other owning and operating costs, and whether the office areas would be heated and air conditioned from the central refrigeration plant. Capital expenditures included all costs of furnishing and installing the insulation and refrigeration equipment. Optimum thickness and type of insulation for each of the refrigerated rooms was calculated by computer from the expected outdoor temperatures of the area as determined from weather records, and costs were determined. Expanded polystyrene proved most economical for rooms above 32° F., and fibrous glass was most economical for rooms below 25°. For rooms with temperatures between 32° and 25°, either insulation may be used. Owning costs were based on yearly estimates for taxes, insurance, and amortization of capital at 6 percent, over 20 years for the central systems and 10 years for the separate systems. Operating costs considered were maintenance of equipment and insulation, and electric power costs.

The system of one central plant for four buildings could be installed for an initial capital expenditure of $893,277, including heating and air conditioning for the offices and working spaces needed to maintain the activity of the 34 firms. A separate building would need to be constructed to house the equipment for the refrigeration system. The costs of owning and operating the central system, including electric power costs, would amount to $190,941 per year.

Four central systems, one in each of the buildings, would cost $966,664 to buy and install, including the heating and air conditioning for the other spaces. A room for the refrigeration equipment would be located under the rear platform of each building. It would cost $239,463 per year to own and operate the four systems.

Individual refrigeration systems for each of the 34 firms would cost $931,598 installed. This figure does not include costs for heating and air conditioning offices and working spaces. These would have to be installed separately at an additional cost of about $72,000 or more, increasing the capital expenditures to 13.4 percent higher than the cost of one central system. The refrigeration units for each firm would be installed in utility tunnels located under the platforms at the rear of the buildings. The cost of owning and operating the 34 systems would amount to $309,157 per year, including electric power.

The weighted temperature-hour approach developed in this study gives an appropriate weighting to the outside temperatures during the year that exceeded the interior design temperature. This procedure was included in the computer program for selecting the optimum thickness of insulation and was also used for calculating the yearly owning and operating costs. Substantial savings result by using this weighted temperature-hour approach.

Detailed sample specifications are given for equipment and building material that will meet the refrigeration needs for such a food distribution center. Methods are shown for adapting the figures given to the requirements of a different climate

[1] Mr. Stahlman resigned from the U.S. Department of Agriculture in March 1969.

INTRODUCTION

[illegible] point toward integrated wholesale food [illegible] facilities for dealers handling different com- [illegible] annually in planning and developing [illegible] to provide facilities for perishable foods. [illegible] for a *reliable*, *efficient*, and *economical* [illegible]

[illegible] or selected on the basis of experience [illegible] optimum plans for cost and effi- [illegible] handling, and conditioning food products in many [illegible] the need for effective marketing. Some facilities [illegible] have occurred in the methods of handling, [illegible] poor arrangement and construction, because of overcrowding; or because of having been built and enlarged with little thought or planning for future operations.

The increase in the per capita consumption of frozen food products and perishable foods requires the food industry to have good facilities in the right locations in order to market high-quality foods at the lowest possible cost. The facilities, equipment, and handling techniques must be planned and developed for efficiency.

In this report, specific data, material, designs, and evaluations have been developed to determine an efficient system for providing the refrigeration requirement in such food distribution centers. Weather data and electrical, labor, and material costs used in this report reflect those current in Chicago at the time this study was made.

BACKGROUND INFORMATION

[illegible] are located in new and modern [illegible] figure 1. These centers provide efficient, [illegible] carloads and truckloads of food from producing [illegible] by truck to retail outlets. The nine [illegible] characterized as multiple-occupancy [illegible] that are completely separated [illegible]

[illegible] developed building plans that will [illegible] for the handling of various commodi- [illegible] buildings. These plans are illustrated in [illegible]

Individual food dealers usually have varying needs for refrigeration to protect the perishable foods they handle. Some foods require subfreezing temperatures, while others require above-freezing temperatures. In many cases, a single dealer will need more than one kind of refrigerated room, each having a different temperature and humidity.

Both individual package systems and centralized systems are used to supply refrigeration to food wholesalers at different locations throughout the country. But reliable scientific information is lacking for comparing the efficiency of operation and costs of installation for these two different types of systems. This has made it difficult to standardize a system that will best suit the needs of modern food distribution centers.

DESCRIPTION OF THE PROBLEM

[illegible] food distribution center in Chicago, Ill., [illegible] effective and economical system of supplying [illegible]

Master Plan of the Food Distribution Center

[illegible] buildings used as the basis for this study. [illegible] buildings which house 34 individual [illegible] and, therefore, require varying amounts [illegible]

Building No. 1 is occupied by 10 fruit-and-vegetable firms, Building No. 2 by eight meat-and-meat-products firms, building No. 3 by 10 poultry-and-egg firms, and Building No. 4 by six grocery firms.

The refrigeration requirements of each dealer and the type of commodity handled are shown in table 1.

In making determinations of cost under this proposal, it is assumed that the basic buildings have been constructed except for the final grading of fill and pouring of floor slabs where floor insulation is required, and that the partitions separating different dealers have been installed. Basic building and partition costs are not considered.

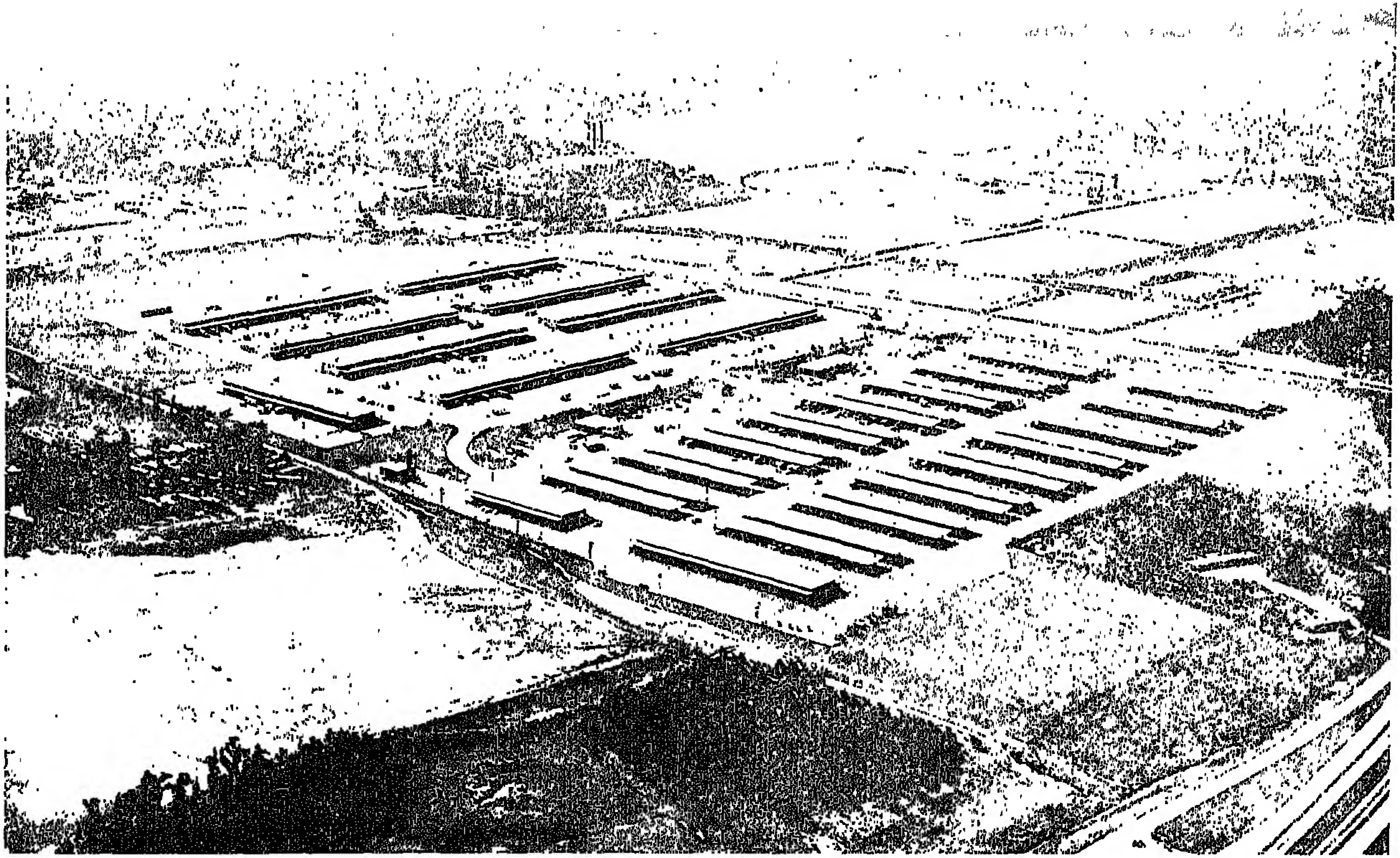

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FIGURE 1.—Aerial view of a modern food distribution center.

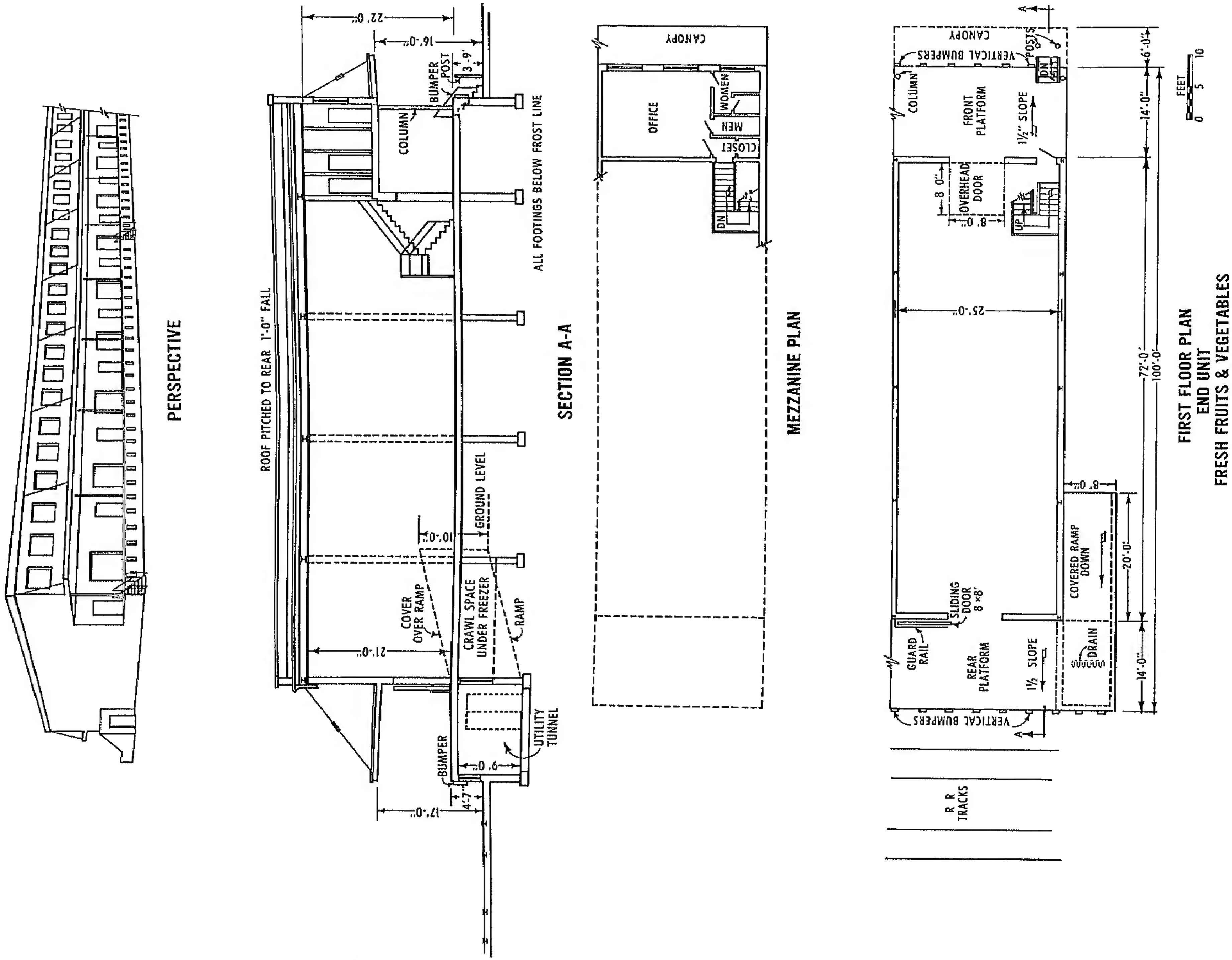
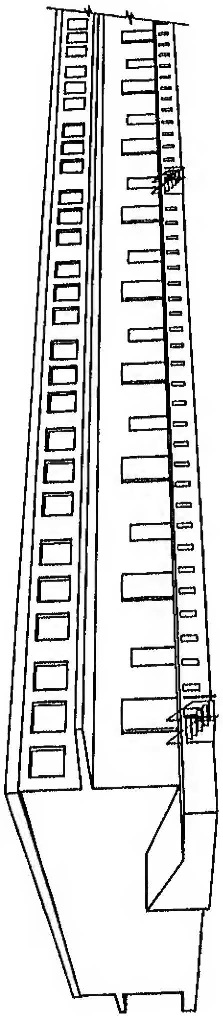
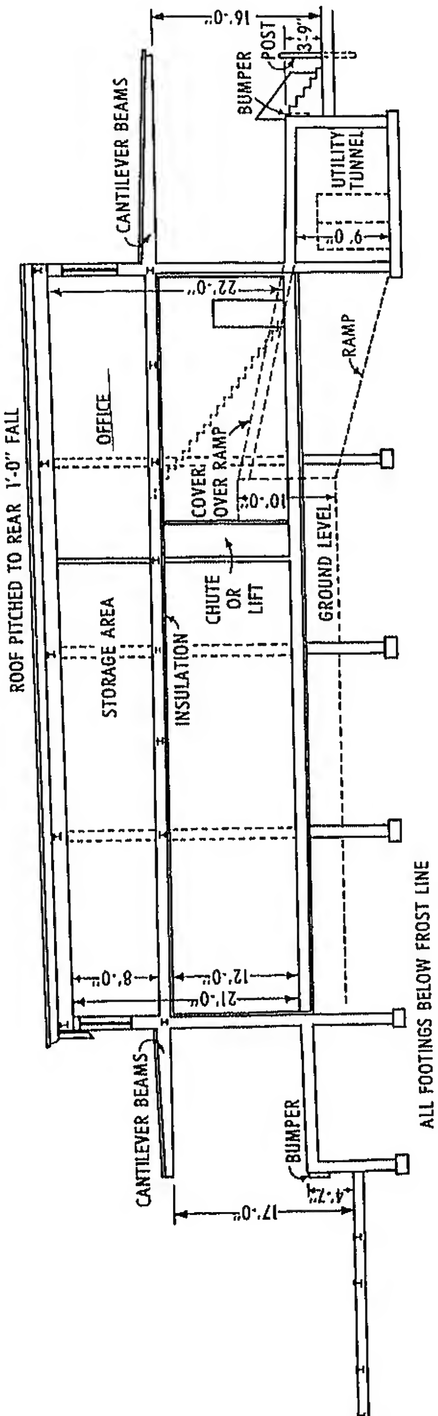


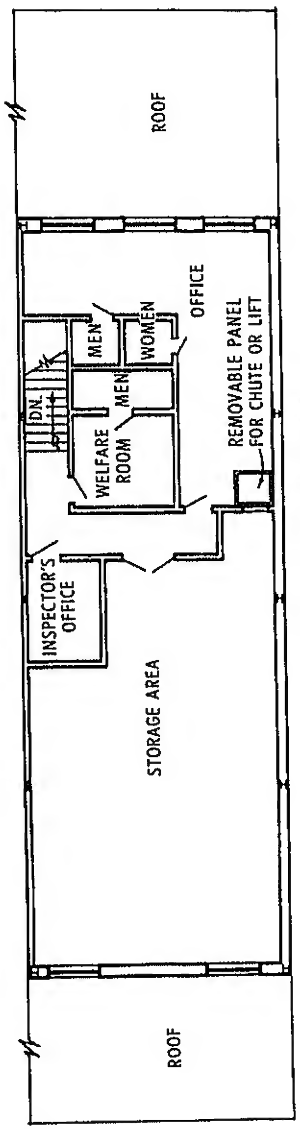
FIGURE 2.—Suggested layout for a fresh-fruit-and-vegetable unit in a multiple-occupancy building.



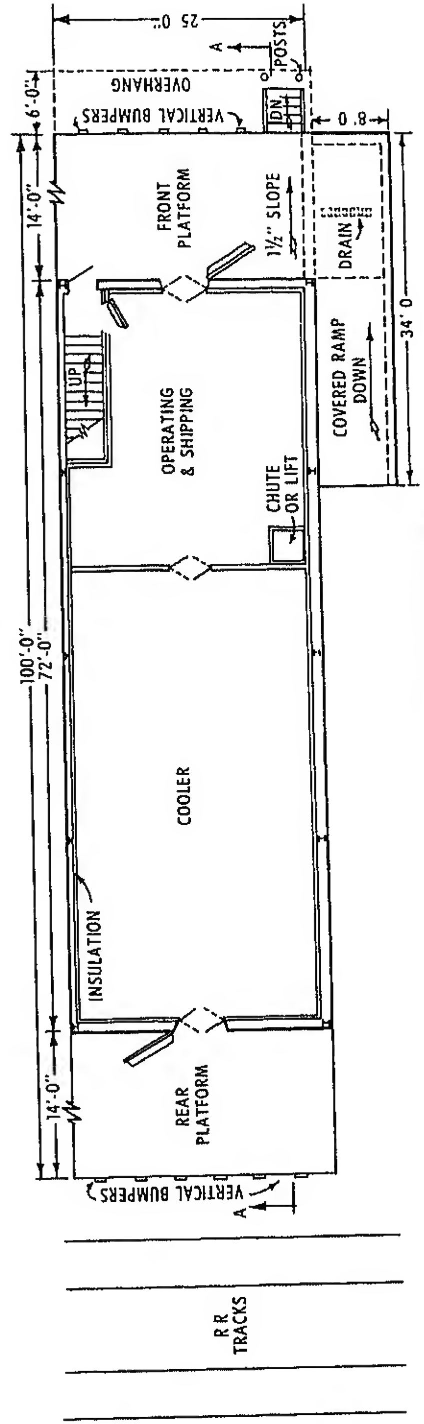
PERSPECTIVE



SECTION A-A



SECOND FLOOR PLAN



FIRST FLOOR PLAN
END UNIT
MEAT & MEAT PRODUCTS

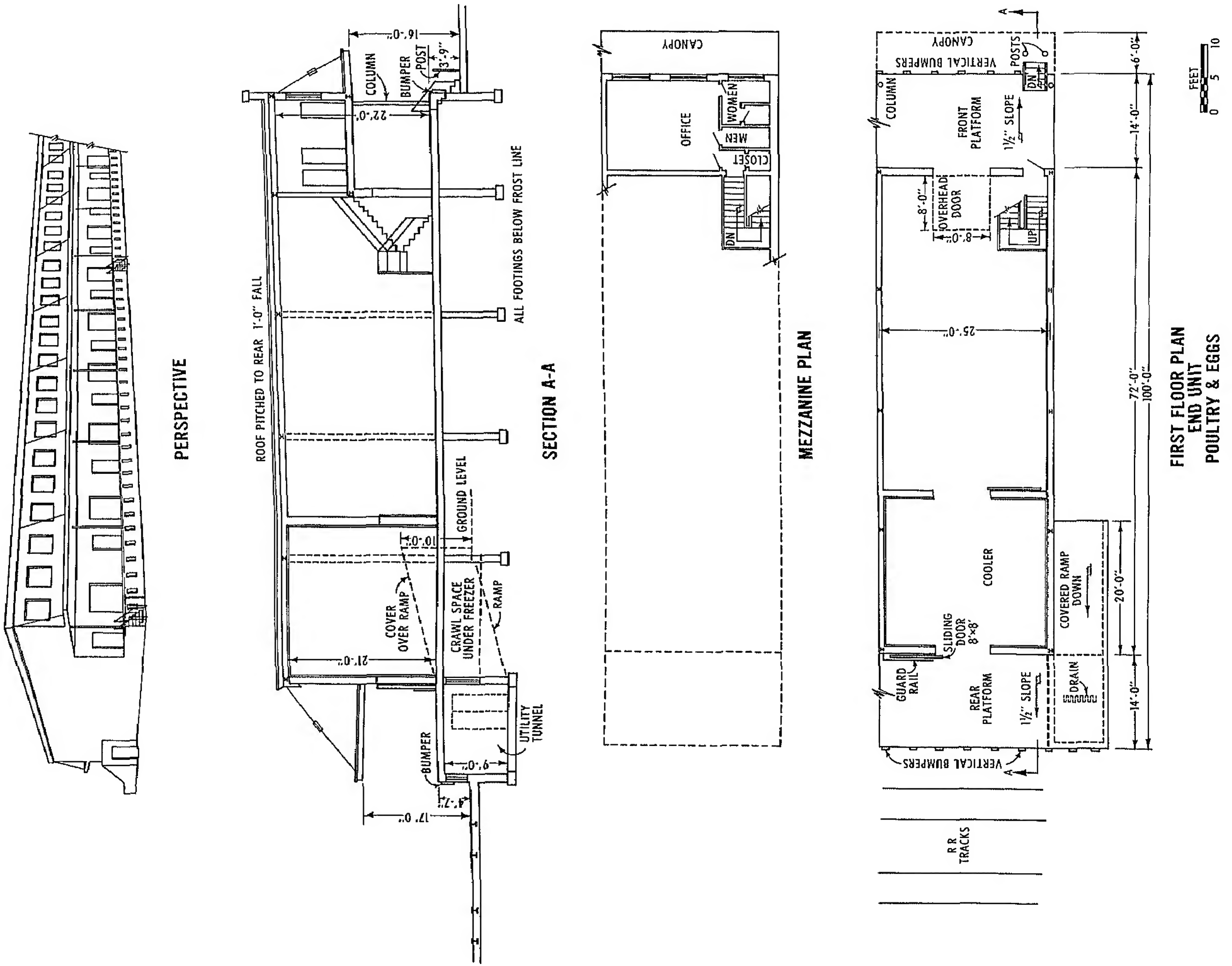


FIGURE 4.—Suggested layout for a poultry-and-egg unit in a multiple-unit ...

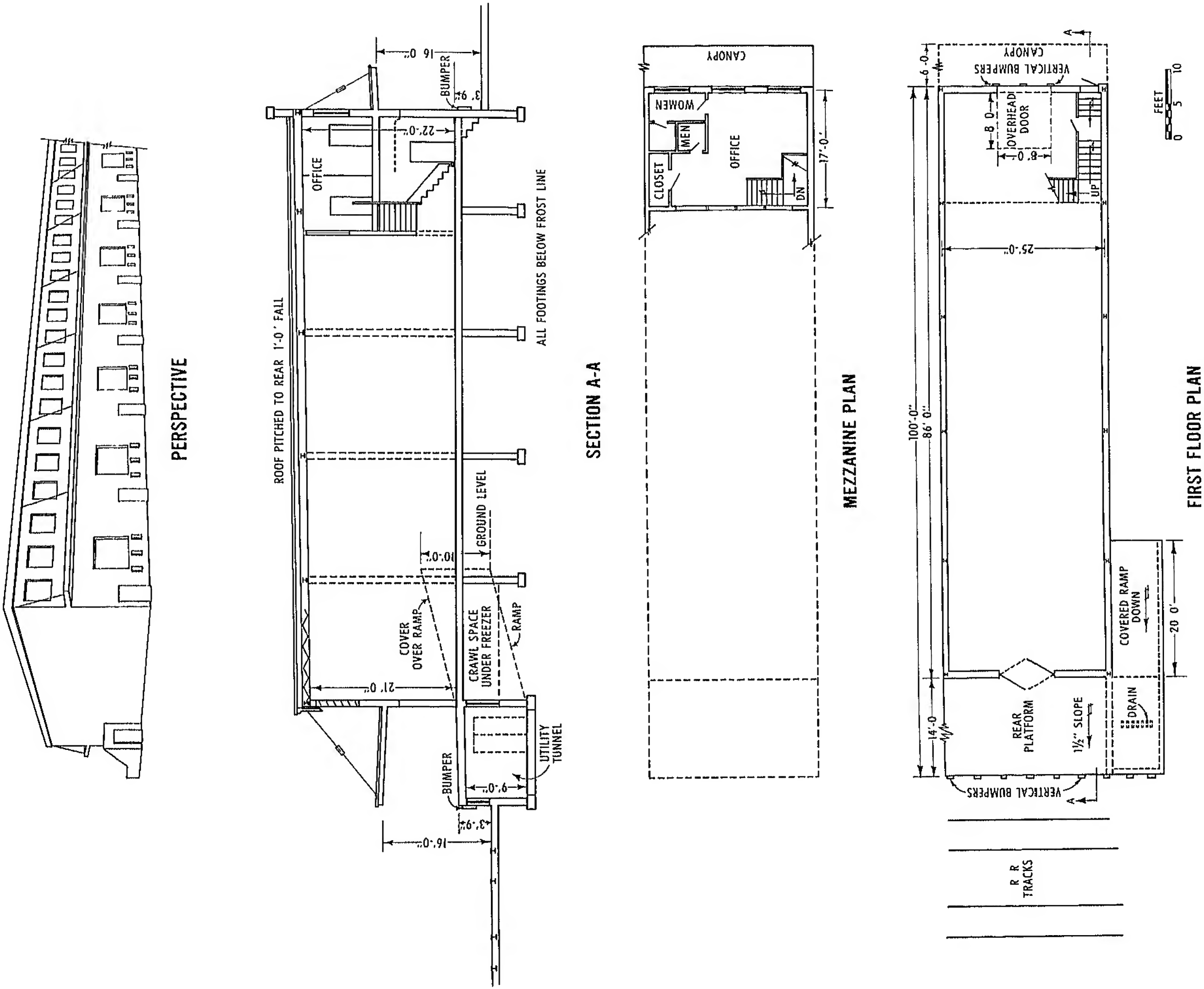


Figure 5.—Suggested layout for a grocery unit in a multiple-occupancy building.

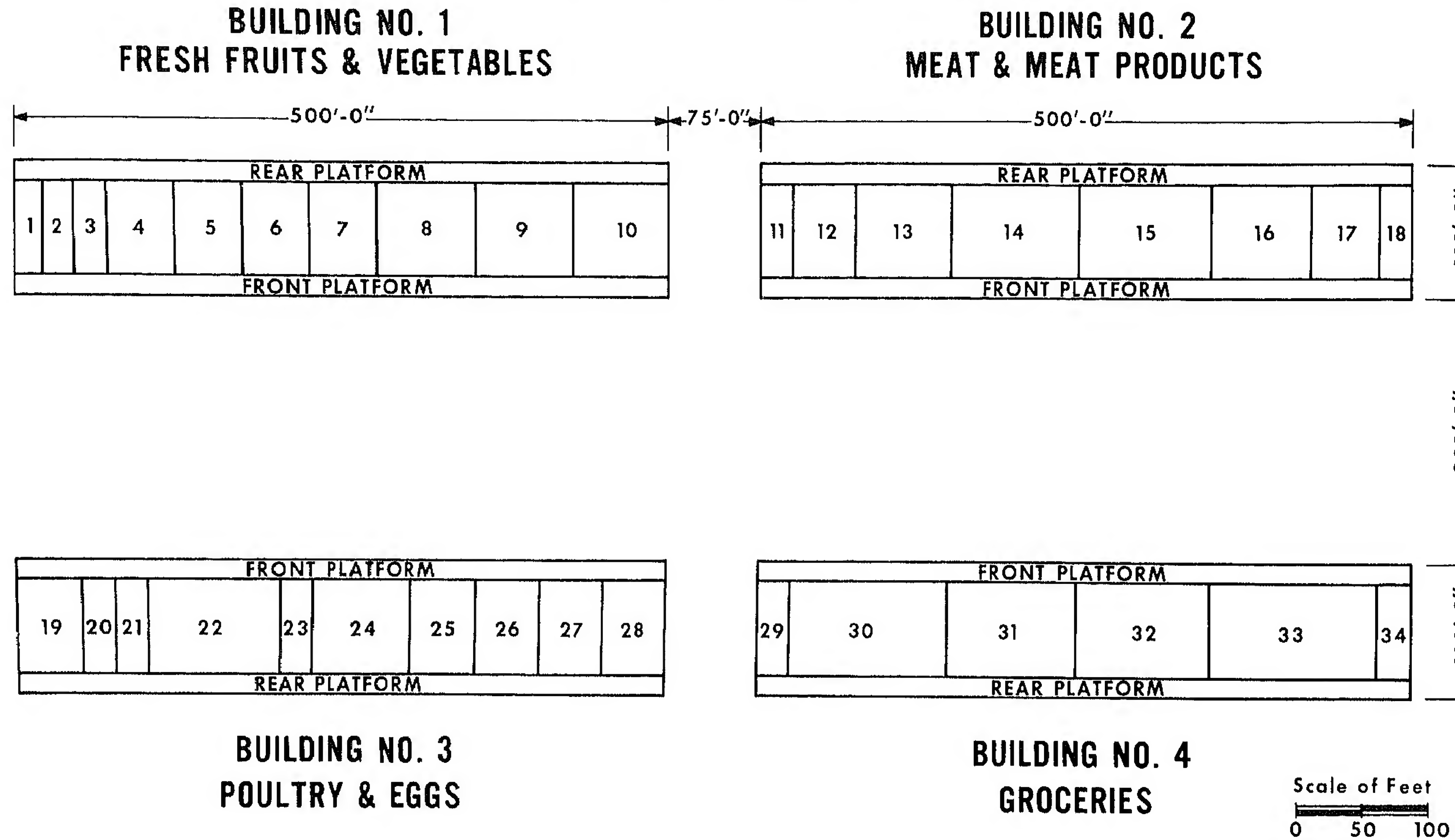


FIGURE 6.—Layout of wholesale food distribution facilities.

TABLE 1.—*Refrigeration requirements for each wholesaler*

Wholesaler number and location	Dimensions and kind of refrigerated space[1]		Temperature and humidity requirements			Kind of product stored in cooler	Condition of product stored in cooler[3]	Additional refrigeration requirements
	Cooler	Freezer	Cooler		Freezer			
			Temperature	Humidity[2]	Temperature			
	Feet	*Feet*	*Degrees F.*	*Percent*	*Degrees F.*			
Building No. 1:								
Fruits and vegetables:								
Firm No. 1	9 × 15 × 10	None	32	90		Fresh fruits and vegetables	Wet	None
Firm No. 2	8 × 10 × 10	do	40	85		do	Dry	Do.
Firm No. 3	24 × 70 × 20	do	45	85		do	Dry	Do.
Firm No. 4	24 × 70 × 20	do	32	85		do	Wet	Do.
Firm No. 5	24 × 34 × 20	do	32	90		do	Wet	Do.
	24 × 35 × 20	do	50	85		do	Dry	Do.
Firm No. 6	49 × 70 × 20	do	50	85		do	Wet	Do.
Firm No. 7	24 × 34 × 20	do	32	90		do	Wet	Do.
	24 × 35 × 20	do	50	85		do	Dry	Do.
	24 × 70 × 20	do	50	85		do	Dry	Do.
Firm No. 8	24 × 44 × 20	do	32	90		do	Wet	Do.
	24 × 25 × 20	do	50	85		do	Dry	Do.
Firm No. 9	74 × 70 × 20	do	50	85		do	Dry	Do.
Firm No. 10	24 × 44 × 20	do	50	85		do	Dry	Do.
	24 × 25 × 20	do	32	90		do	Wet	Do.
Building No. 2:								
Meats and meat products:[4]								
Firm No. 11	24 × 42 × 12	do	32	90		100% carcass meat	Dry	Do.
	24 × 27 × 12	do	50	85		None		Do.
Firm No. 12	42 × 70 × 12	do	32	90		75% carcass; 25% packaged meat.	Dry	Do.
Firm No. 13	74 × 42 × 12	10 × 27 × 12	32	90	−10	75% carcass; 25% packaged meat.	Dry	Do.
	63 × 27 × 12		50	85		None		Do.
Firm No. 14	99 × 42 × 12	20 × 27 × 12	32	85	−10	50 % carcass; 50% packaged meat.	Dry	Do.
	78 × 27 × 12		50	85		None		Do.
Firm No. 15	99 × 42 × 12	30 × 27 × 12	32	90	−10	85% carcass; 15% packaged meat.	Dry	Do.
	68 × 27 × 12		50	85		None		Do.
Firm No. 16	74 × 42 × 12	None	32	90		100% carcass meat	Dry	Do.
	74 × 27 × 12		50	85		None		Do.
Firm No. 17	49 × 42 × 12	10 × 27 × 12	32	85	−10	100% packaged meat	Dry	Do.
	38 × 27 × 12		50	85		None		Do.
Firm No. 18	24 × 42 × 12	10 × 27 × 12	32	85	−10	100% packaged meat	Dry	Do.
	13 × 27 × 12		50	85		None		Do.

See footnotes at end of table on page 10.

TABLE 1.—*Refrigeration requirements for each wholesaler*—Continued

Wholesaler number and location	Dimensions and kind of refrigerated space[1]		Temperature and humidity requirements			Kind of product stored in cooler	Condition of product stored in cooler[3]	Additional refrigeration requireme
	Cooler	Freezer	Cooler		Freezer			
			Temperature	Humidity[2]	Temperature			
Building No. 3	*Feet*	*Feet*	*Degrees F.*	*Percent*	*Degrees F.*			
Poultry and eggs								
Firm No. 19	24 × 35 × 20	24 × 34 × 20	40		−10	Poultry	Wet	Do.
Firm No. 20	24 × 49 × 20	None	40			do	Wet	Do.
Firm No. 21[5]	24 × 25 × 20	8 × 10 × 10	40		−20	do	Wet	12′ × 12′ × 10′ Air-cond. room (7
Firm No. 22[6]	49 × 23 × 20	49 × 22 × 20	40		−10	Poultry	Wet	49′ × 70′ × 20′ Air-cond. room (7
	49 × 23 × 20		50	80		Shell eggs	Dry	
Firm No. 23[7]	24 × 28 × 20	None	50	80		do	Dry	24′ × 41′ × 20′ Air-cond. room (72
Firm No. 24[4]	24 × 50 × 20	24 × 8 × 10	50	80	−20	do	Dry	24′ × 10′ × 10′ Air-cond. room (72
Firm No. 25	None	49 × 49 × 20			−10			None.
Firm No. 26	24 × 27 × 20	49 × 42 × 20	50	80	−10	Shell eggs	Dry	Do
Firm No. 27	24 × 34 × 20	24 × 70 × 20	40		25	Poultry	Wet	Do.
Firm No. 28[6]	49 × 42 × 20	None	40			do	Wet	49′ × 27′ × 20′ Air-cond. room (72
Building No. 4								
Groceries								
Firm No. 29	None	None						None.
Firm No. 30	24 × 34 × 20	24 × 35 × 20	50	85	−10	Fresh fruits and vegetables	Dry	Do.
	24 × 35 × 20		32	90		do	Wet	Do.
Firm No. 31	24 × 35 × 20	None	32	90		do	Wet	Do.
Firm No. 32	None	do						Do.
Firm No. 33	24 × 44 × 20	24 × 25 × 20	32	90	−10	Fresh fruits and vegetables	Wet	Do.
Firm No. 34	None	None						Do.

[1] Dimensions shown are inside dimensions (wall to wall and floor to ceiling) in width, length, and height.
[2] Minimum requirement.
[3] "Wet" indicates product packed in melting ice. "Dry" indicates no excess moisture.
[4] [illegible] work area for cutting up, boning, packaging, and order assembly operations.
[5] Air-conditioned area used for moderate work such as cutting up poultry and egg-breaking operations.
[6] Air-conditioned area used for moderate work such as order assembly.
[7] Air-conditioned area used for egg-grading operations and moderate work such as order assembly.

Three Situations To Be Considered

Three different situations are to be considered in determining the most efficient and reliable refrigeration system for each firm. Each situation applies to the buildings as outlined in figure 6 and the requirements as summarized in table 1. The situations are:

Situation I. Each of the 34 dealers provides, operates, and maintains his own individ refrigeration system or systems.

Situation II. One central system provides refrigeration, including air conditioning and heati to all 34 dealers.

Situation III. One separate, central refrigeration system in each of the four buildings suppl the refrigeration needed, including air conditioning and heating, to its respecti building.

Data and Questions To Be Analyzed

Each situation is to be thoroughly analyzed and the final calculations and selections, as applicable, made for:

(a) The most efficient type, optimum thickness, and total installed costs of insulation required in each refrigerated room, including the air-conditioned work areas. The cost includes inside finishes over the insulation for all walls and ceilings.

(b) The total refrigeration tonnage required by each dealer, by each of the four buildings, and by all four buildings.

(c) The necessary equipment, properly selected and located, to meet the tonnage requirements in (b) above, and the temperature and humidity requirements as listed in table 1.

(d) The layout and arrangement of the equipment room(s) as required for Situations II and III.

(e) The best location for the central refrigeration plant for Situation II in relation to the four multiple-occupancy buildings.

(f) The initial cost for the installed refrigeration equipment as selected in (c) above.

(g) The most efficient type of refrigerant to use with the equipment selected in (c) above.

(h) The average annual cost of owning and operating the individual or central refrigeration systems. This cost is to include the individual costs for operation, depreciation, maintenance, taxes, and insurance.

(i) The air-conditioning and heating equipment to handle the office areas in Situations II and III, based on a method of supplying the required heating and cooling from the central refrigeration equipment.

METHODS USED TO DEVELOP COSTS

Two classifications of costs are to be considered for each situation—the "initial capital expenditures" and the "annual owning and operating costs." In Situation I, these costs are broken down by firm, since each firm has its own individual package system. In Situation II, the costs are calculated by building, and the four building totals are summarized into a complete cost for one central system. For Situation III, the costs are calculated by building since each building has its own central system.

The two cost classifications are made up of various costs as covered in this section, and are applied to all three situations. All cost figures are included in the individual sections covering each situation.

The installation costs are based on labor rates in the Chicago, Ill. area in 1965.

Initial Capital Expenditures

Refrigeration equipment

The installed costs include labor and materials for all air-handling units, condensing units, central-system equipment, piping, piping insulation, and controls.

Insulation

The installed costs include labor and materials for all cold-room insulations. The installed cost in \$/ft.^{2} is determined for each type and thickness of insulation as used per situation. The total cost is found by multiplying the area of a surface by the installed cost. If a firm has more than one refrigerated room, the total cost is the sum of the individual room costs.

Cold-storage doors

The types of doors used and the installed costs are listed in the section on cold-storage door specifications.

Air-conditioning equipment

All buildings in Situation I are assumed to be *without* air conditioning. The installed-cost figures for air-conditioning equipment for the other two situations are developed in "Air Conditioning-Heating, for Situations II and III." Air-conditioning components are listed in the bills of materials for central-system equipment rooms, where they are identified by a footnote reference.

Refrigerant metering devices

Refrigerant metering devices are used in Situations II and III only. The types of metering devices are specified in the typical specifications, and the installed costs are listed in the summary cost tables for each situation.

Annual Owning and Operating Costs

Amortization of capital expenditures

The "Equal Annual Payment" method[2] is used, consisting of a dollars-per-year owning and operating cost, set up in an annuity form so that at the end of a 20-

[2] See Marks' Mechanical Engineers' Handbook, 6th ed., ch. 17, p. 44.

year period,[3] the initial investment will be returned, with interest compounded annually. The "Equal Annual Payment Rate" is determined by the formula

$$x = \frac{i\,(1+i)^n}{(1+i)^n - 1}$$

where x = equal annual payment rate factor
i = interest rate in percent ÷ 100
n = depreciation time in years

For the 20-year equipment life and 6-percent interest rate[4] as used in this study

$$x = \frac{.06\,(1+.06)^{20}}{(1+.06)^{20} - 1} = 0.08718$$

Maintenance of insulation

The cost for maintaining insulation is quite small. It was arbitrarily taken at a fixed value of 2 percent of the insulation cost.

Maintenance of refrigeration equipment

For unitary package systems, as in Situation I, the cost varies with the type of system installed. An average value of 10 percent of the refrigeration-equipment cost was arbitrarily selected.

For central-system installations, as in Situations II and III, the maintenance is based on the cost of a manufacturer's certified contract, which includes inspections, preventive maintenance, oil, refrigerant, replacement parts, labor, and 24-hour emergency service. A minimum of one thorough inspection per month is included.

Maintenance of air-conditioning equipment

This maintenance is calculated on the basis of a contract similar to that decribed for the refrigeration equipment above.

Insurance

The insurance rate is based on 80-percent evaluation. It includes fire and extended coverage at a base rate of \$2.26/\$1,000. The adjusted value for 80-percent evaluation is equal to 0.80 × \$2.26/\$1,000 = \$1.81/\$1,000.

Taxes

The tax rate is based on 65-percent evaluation times the Illinois State correction factor of 1.42 and tax rate of \$5.284/\$1,000. The adjusted value for 65-percent evaluation is equal to \$5.284/\$1,000 × 1.42 × 0.65 = \$4.88/\$1,000.

Electric power cost

The electrical rates are based on figures supplied by the Commonwealth Edison Company of Chicago, Ill. The total power cost includes a "demand" cost and an "energy" cost. The formulas, methods, and base cost figures used to calculate the power cost are shown in the section on cost comparisons.

SELECTING THE INSULATION

Types of Insulation Material

The following basic types of insulation material were considered for this study:

- Expanded polystyrene board
- Extruded polystyrene board
- Corkboard
- Polyurethane board
- Fibrous glass
- Cellular glass

Figure 7 compares the thermal conductivity of these types of insulation at various temperature differences.[5]

In this study the board types of insulation are considered as being installed mechanically with studs that allow for thermal expansion of the exterior wall without destroying the vapor barrier. The fibrous glass is assumed to be installed with nonconducting studs and with a hard fibrous-glass board finish. The cellular glass is considered freestanding and, like the board insulations, is finished with ½ inch of cement applied in two coats.

Maintaining a good vapor barrier with adhesive-type installations is a common problem. A 10-mil polyethylene sheet is added as a vapor barrier on the warm side of all insulations considered here.

Blown-on foams, such as polyurethane, are not included in this study because there has not been enough proven experience with these materials. Atmospheric conditions are very important in this type of installation. All major manufacturers feel that foamed-in-place insulations are fine where atmospheric conditions are controlled; but that they are not to be recommended for general field work because the weather, availability of skilled labor, and quality of equipment introduce numerous variables. Other problems concerning adherence to vapor barriers and finished surfaces make the use of such foams questionable, at present, for cold-storage applications.

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[3] A 20-year amortization period is used for all insulation and refrigeration equipment, except in Situation I, where a 10-year period is used for the refrigeration equipment. Package equipment, on the average, has a shorter life than heavy central-system equipment.

[4] No attempt was made to reflect accurate interest rates that might exist at any given time. Six percent was selected merely to demonstrate the application and effect of interest costs.

[5] Determined from the published data of the major insulation manufacturers. Thermal conductivity is measured in B.t.u./hr/-ft.²-° F./in.

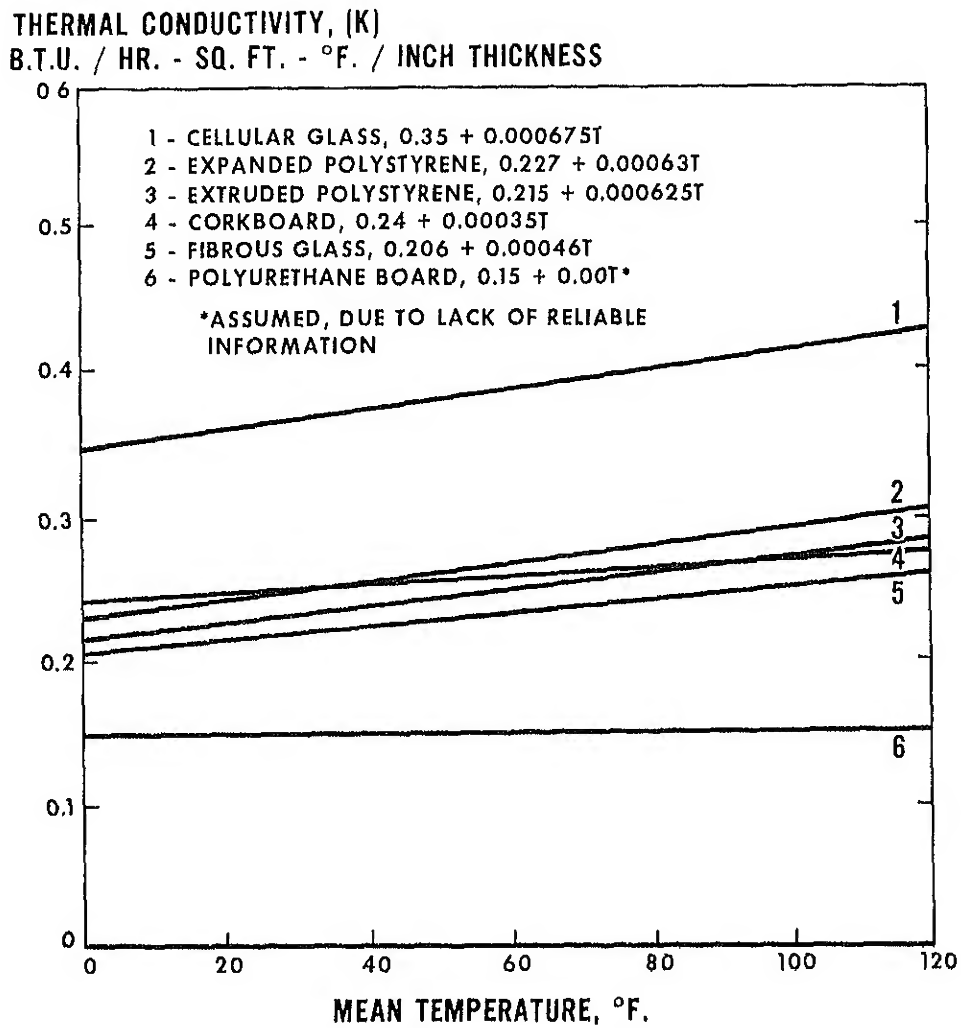

FIGURE 7.—Thermal conductivity of various insulating materials at different temperatures.

The Optimum Balance Between Insulation and Refrigeration Costs

Research into the economic phase of designing cold-storage rooms has revealed the inadequacy of conventional design methods. Customary practice begins by selecting a thickness of insulation material as recommended by a manufacturer's catalog. The selection of insulation thickness has been followed by load calculations and the selection of a refrigeration system based on an outdoor design temperature, usually 95° F. dry bulb, with no apparent attempt to arrive at a balance between insulation costs and refrigeration costs.

To select insulation properly for a refrigerated room, an economic analysis is needed to determine an optimum balance between insulation and refrigeration costs. Extensive research has not uncovered a study in which a practical approach has been made toward arriving at such an optimum balance, though several theoretical studies show that as insulation thickness increases, insulation cost increases and the cost of required refrigeration decreases; and vice versa.

A graph can be drawn to determine the optimum balance point by plotting individual curves for the total insulation and refrigeration costs. These two curves are added to obtain a total cost curve. The low point on the total cost curve indicates the optimum balance point (fig. 8). The thickness of the insulation at this point is the optimum insulation thickness.

One of the early papers on optimum thickness of insulation was "The Economic Thickness of Insulation in the Refrigeration Field," by P. Nicholls (*Refrigerating Engineering* 9 (5): 152, Nov. 1922). This paper discussed several cost items to be included and gave an equation to calculate insulation thickness for a minimum cost.

L. B. McMillan, in the 1926 *Transactions* of the American Society of Mechanical Engineers, discussed several aspects of heat transfer through insulation at moderate and high temperatures and derived an equation for calculating the optimum thickness of insulation for flat surfaces. However, he assumed a single lump-sum factor for the cost of heat and another single factor for the insulation costs.

In 1960, the Union Carbide Corp., Charleston, W.Va., sponsored a project of the Engineering Experimental Station of West Virginia University to develop a "Manual on Economic Thickness of Insulation for Flat Surfaces and Pipes." Development of data for the manual followed the general procedure given by McMillan in 1926. It differed from McMillan's method, however, in the determination of the base insulation-cost factor and the method of accounting for maintenance and capital recovery (amortization) factors in the "owning and operating" costs.

About 1962, the National Insulation Manufacturers Association (NIMA) reprinted the West Virginia University tables, with some slight additional information, as their official manual on Economic Thickness of Insulation for hot pipes and surfaces losing heat to the air around them.

In a bulletin on "Cold Storage Systems"; put out by the Owens-Corning Fiberglas Corp. in June 1965, one section is entitled "Economic Thickness." The equations and procedures used are based on the NIMA Manual. However, the cost factors given in the manual were converted to units more applicable to refrigeration systems. In addition, an equation for "Capital Cost Recovery," using factors of "depreciation period" and "interest rate," is included.

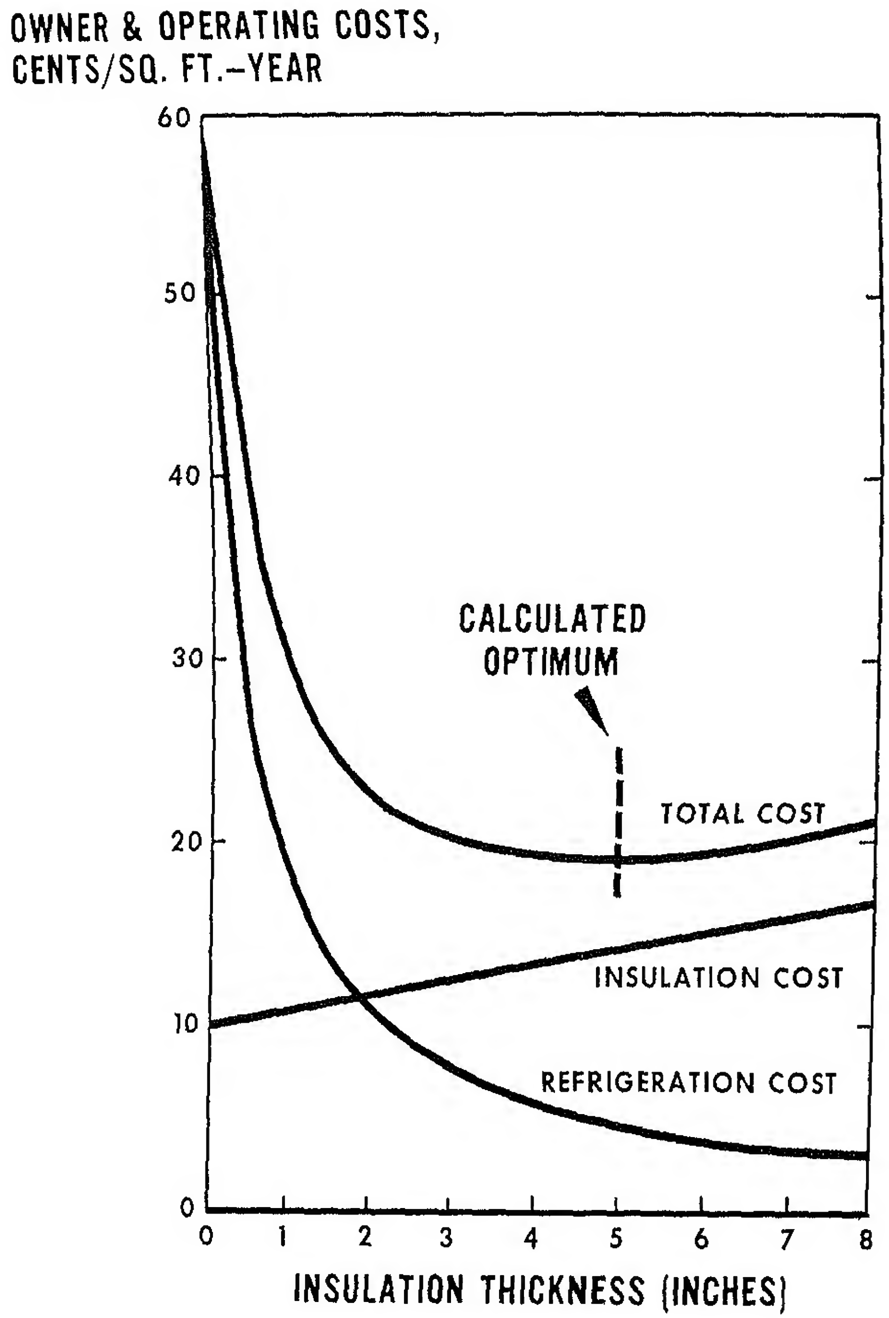

FIGURE 8.—Optimum thickness for insulation.

How the Computer Is Used to Analyze and Select the Optimum Thickness of Insulation

It would be a time-consuming task to calculate by hand the optimum thickn of insulation for all the possible combinations of interior and exterior temperatur types of insulation, changes in thermal conductivity at the various temperat differences, and the varying costs for investment in and operation of the ma mechanical refrigeration systems. In contrast, the computer effectively perfo similar calculations with high speed and accuracy.

It appeared that the basic direct solution for economic thickness of insulat proposed by L. B. McMillan was rather widely accepted. To see if it could adapted to computer programming, the data used in the West Virginia Univer study and the fiberglass company's bulletin were programmed into the compu to determine the total owning-operating cost curves. These results were chec by hand calculations.

The computer-derived optimums differed from the hand-calculated optimums a thickness of 0.3 inch to 0.7 inch, either higher or lower. The exact reason for difference is not known. Perhaps the minimum point on the curve (fig. 8) is n sharp one, and the slope deviation from zero is within the accuracy of the comp tional method. Also, one of the base constants in the equations supplied by V Virginia University reportedly was calculated as the average value from a grou data having a fair spread.

The computer starts the calculations with the insulation thickness given in input data and calculates the corresponding sum of insulation and refrigera owning-operating costs. It then increases the insulation thickness considere 1.0-inch increments as long as the new cost is less than the previous value. W the new cost becomes greater than the previous value, the change in thick reverses to a decrease by 0.1-inch increments. The decrease again continues as as the cost is greater than the previous value. At the "change point" the incre reverses again to an increase, by steps of 0.01 inch. When the change poi reached this time, the insulation is considered to have reached the optimum t ness within a reasonable accuracy. If the difference between successive valu 0.00000000 at any time during the above computation, the program immedi jumps to the assumption that it has reached the optimum thickness of insula without going any further.

Solutions obtained by this method result in odd decimal thicknesses, wh insulation is normally available in ½-inch increments. In cases where the de is 0.10 inch or less, the program reverts to the lesser ½-inch increment; bu tween 0.10 inch and 0.49 inch, the program moves up to the next ½-inch incre The result is known as the "commercial" optimum. The heat transfer and factors for this commercial optimum thickness are then calculated and printe This rounding off is weighted on the high side, since the cost curves are gen flatter as the thickness increases. The exact division point is an arbitrary on can be changed if desired.

Assumed refrigeration costs as computer input data

To select an optimum thickness of insulation, a computer must have data on the type of refrigeration system, the initial capital expenditures, and the owning and operating costs. Actual refrigeration costs cannot be used as input to the computer, however, because the actual costs are derived only as a result of the computations, so they cannot be known in advance. It is necessary, therefore, to estimate or assume these costs (table 2.)

TABLE 2.—*Assumed refrigeration costs used as computer input data for the different situations*

Situation and room temperature (° F.)	Assumed cost		
	Refrigeration equipment	Operation	Maintenance
	Dollars[1]	*Dollars*[2]	*Percentage*[3]
Situation I.			
40° and above	750.00	0.615	10.0
25° to 34°	1,100.00	.742	10.0
−20° to −10°	1,250.00	1.350	10.0
Situation II:			
25° and above	725.00	.168	5.7
−20° to −10°	1,200.00	.259	5.7
Situation III:			
20° and above	825.00	.253	8.2
−20° to −10°	1,400.00	.390	3.2

[1] Dollars per ton of refrigeration ($/TR).
[2] $/TR per 24 hours of operation.
[3] Percentage of capital cost per year.

The assumed costs used in this study are accepted figures within the refrigeration industry. While the computations were being made, the contractor conducted research to ascertain that the figures being used as input data were realistic. When the final figures on actual refrigeration costs were compared to the assumed values listed in table 2, the assumed values proved to have been realistic.

Insulation cost figures developed for input data

Insulation cost figures are assembled on an installed basis from data supplied by the major manufacturers and insulation contractors. Basically, the cost information programmed is of two types: (1) a fixed price per *board foot* of insulation material and (2) a fixed price per *square foot* of surface area, which includes such items as installation labor, studs, vapor barrier, ledger strips, interior finish, and ceiling supports. Labor cost is figured on the basis of cost/sq. ft./layer of insulation. Two layers are used on all rooms designed for a temperature of 32° F. and below. Chicago labor rates for 1965 are used.

The final installed-cost figures for the various insulations include the insulation material as well as the fixed price per *square foot* of surface area, as described above.

The cost of money—interest rates considered

For purposes of this study, the method used for amortization of capital expenditures is the "equal annual payment" method. By this method, a dollars-per-year owning and operating cost is set up in an annuity form so that by the end of a 20-year period the initial investment will be returned, with interest compounded annually. A 10-year period was used for the unitary package systems in Situation I. A high salvage value will be realized at the end of the amortization period, because money was provided annually for maintenance.

A 6-percent interest rate is used in this study. No attempt was made to reflect accurate interest rates that might exist at any given time. One computer series was run, using a 10-percent interest rate, to determine what effect the cost of money would have on results. The difference in interest rates, as expected, made little difference in the selection of the thickness of insulation. If a 10-percent rate were to be anticipated, however, the thickness of the insulation would drop one-half inch in walls with up to 6 inches of insulation, and 1 inch in walls with insulation thicknesses of more than 6 inches.

Figure 9 illustrates the difference in owning and operating costs attributed to wall insulation when a 6-percent or a 10-percent interest rate on the amortization allowance is used. More important, it also illustrates the difference between using the *conventional design* approach and a *weighted-temperature-hour approach* as devised for this study to select wall, floor, and ceiling insulations; and to calculate operating costs.

Figure 9 cannot be interpreted as a smooth curve, because the points between −10° and 25° F. have not been calculated. No rooms within this food distribution center complex were designed for these intermediate temperatures.

Temperature profile of Chicago, Illinois

Figure 10 illustrates the temperature profile of the Chicago, Ill. area, based on U.S. Weather Bureau data collected hourly during 1965 at Midway Airport.

To simplify the input data to the computer program, 10° F. increments are used. For example, all hours in which the temperatures were in the 90's are considered as 95°, in the 80's as 85°, and in the 70's as 75° F. Detailed checks on this method revealed that a variance in mean temperature of less than ±1° existed on any day, and that a variance of less than ±1 percent existed on any annual degree-hour basis. Figure 10 shows the total number of hours during 1965 in which the outdoor temperature exceeded the design storage-room temperature.

The temperature profile in figure 10 is used extensively in developing the weighted-temperature-hour approach for all wall surfaces. A temperature penalty

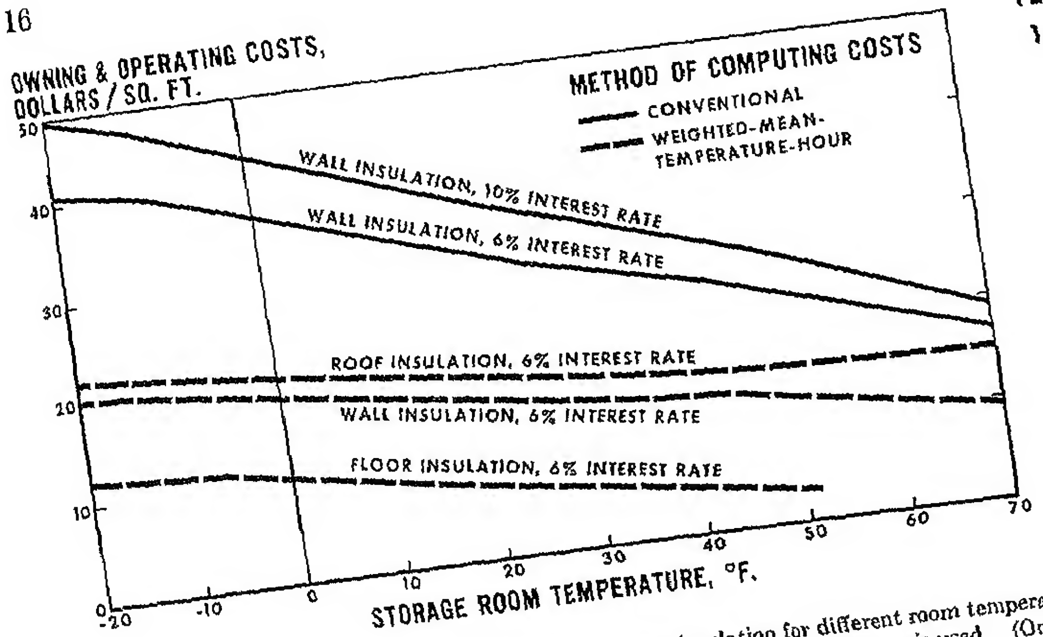

FIGURE 9.—Owning and operating costs attributed to wall insulation for different room temperatures when a 6-percent or a 10-percent interest rate on depreciation allowances is used. (One central system).

for the added heat load caused by direct sunlight on the roof surfaces must be included.

The weighted-temperature-hour approach developed for this study

The conventional method of designing a cold-storage room uses a 95° F. outside dry-bulb temperature as a basis for selecting the insulation thickness and refrigeration equipment. Insulation and equipment are selected for the maximum load that might exist at any one time. This maximum would occur with a full product load and a 95° outside dry-bulb temperature. When the outside temperature is lower than 95° and the product load is less than 100 percent, the refrigeration equipment will be running at part capacity.

Figure 10 shows, however, that the temperature in Chicago was in the 95° F. range for a total of only 36 hours during 1965. Why select insulation and calculate operating costs for a design temperature that exists for less than 0.5 percent of the time? Design considerations would be more realistic if the approach emphasized the number of hours during the year that the outside temperature exceeds the interior design temperature.

Such an approach, called the weighted-temperature-hour approach, was developed for this study. This approach gives an appropriate weighting to the temperatures during the year that exceed an assumed interior temperature.

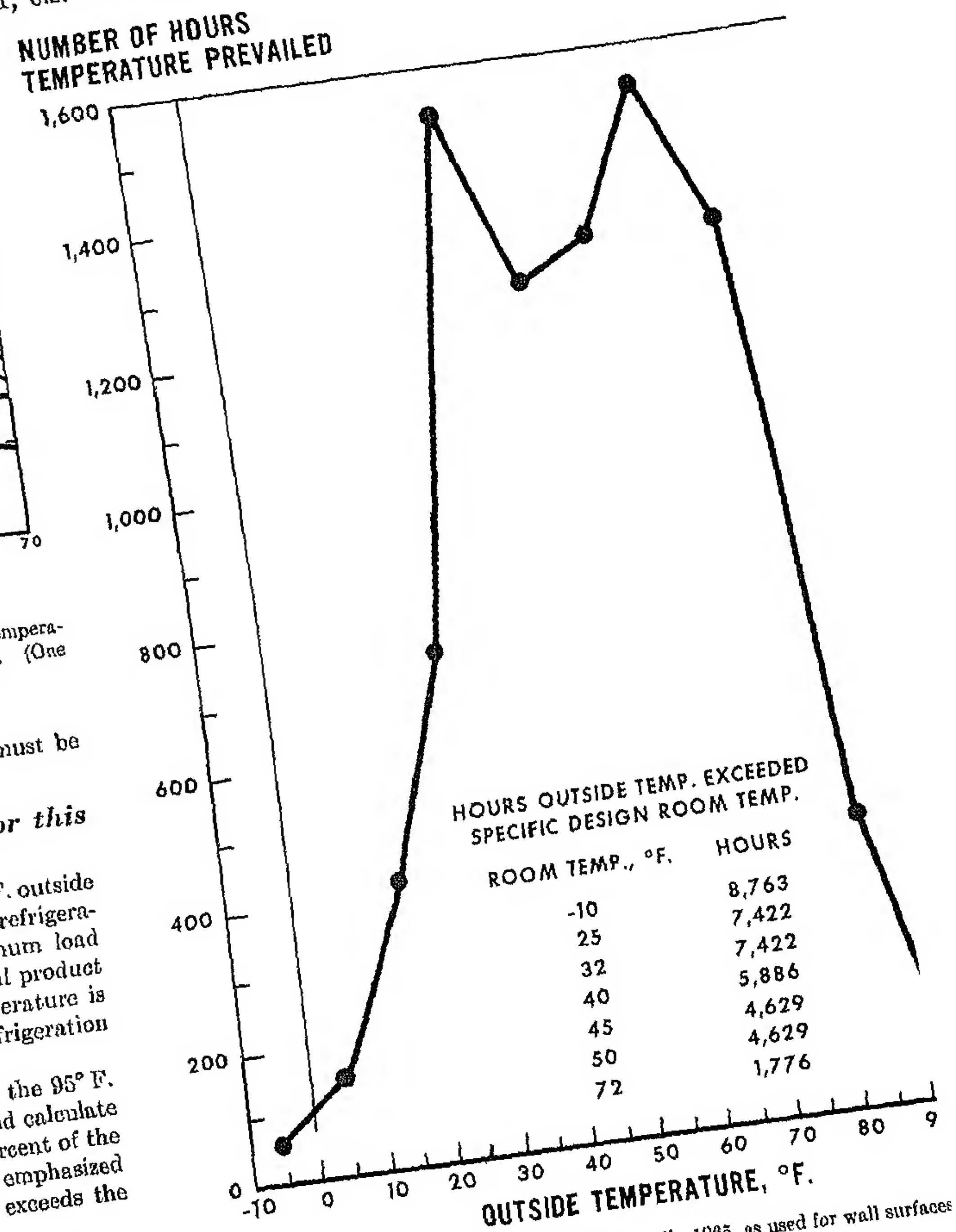

FIGURE 10.—Temperature profile of Chicago, Ill., 1965, as used for wall surfaces

Weighted temperature difference is found by: (1) Multiplying each of the differences between the outside and the storage room air dry-bulb temperatures by the respective cumulative number of hours that each temperature difference exists, (2) adding the products of the multiplication, then (3) dividing the result by the sum of the cumulative hours of the temperature differences.

An example of the procedure for finding weighted temperature difference (WTD) is illustrated by the formula below. The formula assumes a 72° F. storage room in Chicago, and uses the outside temperatures during the year that exceeded the interior design temperature, along with the number of cumulative hours of such temperatures, from figure 10.

$$\mathrm{WTD} = \frac{\sum[(\Delta t_1 \times \mathrm{Hr.}_1) + (\Delta t_2 \times \mathrm{Hr.}_2) + (\Delta t_3 \times \mathrm{Hr.}_3)]}{(\mathrm{Hr.}_1 + \mathrm{Hr.}_2 + \mathrm{Hr.}_3)}$$

where:

WTD = weighted temperature difference (° F.)
$\sum$ = summation of
Δt = difference between two air temperatures
Δt_1 = 95° − 72° = 23° F.
Δt_2 = 85° − 72° = 13° F.
Δt_3 = 75° − 72° = 3° F.
Hr. = A cumulative period of measured time
$\mathrm{Hr.}_1$ = 36 hours
$\mathrm{Hr.}_2$ = 413 hours
$\mathrm{Hr.}_3$ = 1,327 hours

$$\mathrm{WTD} = \frac{(23°\text{ F.} \times 36\text{ hrs.}) + (13°\text{ F.} \times 413\text{ hrs.}) + (3°\text{ F.} \times 1{,}327\text{ hrs.})}{(36\text{ hrs.} + 413\text{ hrs.} + 1{,}327\text{ hrs.})}$$

$$= 5.7°\text{ F.}$$

For this particular example, the weighted temperature difference of 5.7° F. across an outside wall would be used to select the proper insulation thickness. Equipment would be selected by using the standard design temperature of 95° (minus the room temperature of 72°, equals 23°) to calculate the transmission-heat gain through the wall. If the 23° temperature difference were used to select the insulation, a much thicker insulation would be indicated.

The same type of calculation can be made for other storage-room temperatures, and in cities other than Chicago.

Figure 9 shows the important difference between using the conventional design approach and the weighted-temperature-hour approach to determine insulation requirements. The weighted-temperature-hour approach saves 57 percent of the owning and operating costs on 72° F. rooms, 32 percent on 40° rooms, 25 percent on 32° rooms, and 9 percent on −10° rooms.[6] On the basis of these results, it was decided that the most economical insulation thicknesses would be selected on the basis of a 6-percent return on investment, and using the weighted-temperature-hour approach.

[6] The percentages in savings are not taken directly from figure 9. Figure 9 illustrates only the wall portion of the information required to calculate the overall savings.

Thermal design criteria for building construction

Inside walls.—A thermal resistance of 2.5 hr.-ft.²-° F./B.t.u. is used for the inside walls, excluding the insulation resistance. The insulation is selected on the basis that all areas will be maintained at the design conditions and that all unconditioned areas will have an average temperature of 80° during the year. All insulation is installed on the cold side of the wall. Figure 11 shows detailed drawings of typical wall construction.

Outside walls.—A thermal resistance of 2.22 hr.-ft.²-° F./B.t.u. is used for outside walls, excluding the insulation resistance.

Floors.—Floor insulation is selected on the basis of an average ground temperature of 55° F. throughout. A thermal resistance of 2.16 hr.-ft.²-° F./B.t.u. is used for the floors, excluding the insulation resistance. Floor insulation is not recommended when the owning and operating cost for refrigeration alone is less than the combined cost of insulation plus refrigeration. Figure 12 is a detail drawing of a typical floor installation.

Ceilings or roofs.—The basis for the selection of ceiling or roof insulation is much the same as for wall insulation, except that the calculation of the hourly difference between the storage-room temperature and the outside temperature includes the addition of a penalty of 45° added to the outside temperature to correct for the sun load, or extra heat caused by direct sunlight. Sun data for 1965 was obtained from the U.S. Weather Bureau data at Midway Airport.

A thermal resistance of 1.95 hr.-ft.²-° F./B.t.u. is used for the ceiling structure, excluding the insulation resistance. Figure 13 shows a typical freezer-ceiling installation.

Final insulation selections for all situations

Tables 3, 4, 5, 6, 7, and 8 list the final insulation selections, based on computer runs, for each of the situations considered within this report.

These tables give the heat gain per square foot of surface area (Q/A) in B.t.u./hr.-ft.², using the weighted-temperature difference (WTD Q/A) and the conventional 95° F. outside dry-bulb design temperature (Standard Q/A).

The Standard Q/A column under the ceiling-insulation schedule for each situation includes a 20° F. penalty added to the conventional 95° for calculating the transmission-heat gain used in selecting equipment. This penalty is not to be confused with the 45° penalty that is added for sun load when calculating the weighted temperature difference between the ceiling or roof and the storage room as used for selecting the proper thickness of insulation.

Code letters are used on the individual building drawings for the different temperature conditions and the thicknesses and types of insulation selected to meet the particular requirement. Code letters "A" through "H" inclusive designate outside walls, and letters "I" through "P-3" designate inside walls. The listings in tables 3, 5, and 7 indicate that thicker insulation is required on the inside wall of a 50° F. room next to an unrefrigerated space, where the heat load exists for

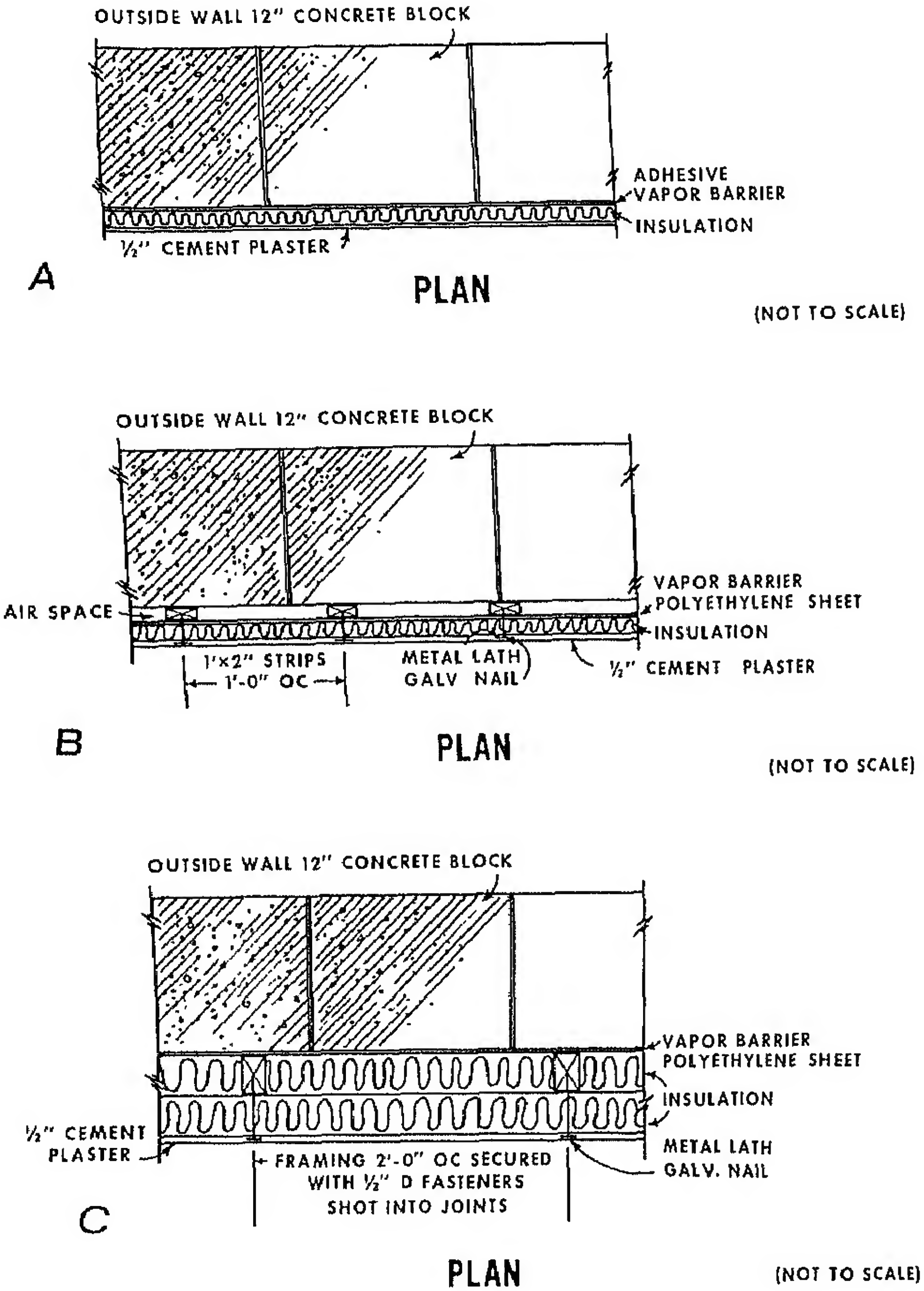

FIGURE 11.—*A*. Details of typical wall insulation and construction—72° F. rooms. *B*. Details of typical wall insulation and construction—40° F. to 72° F rooms. *C* Details of typical wall insulation and construction—32° F. rooms and below.

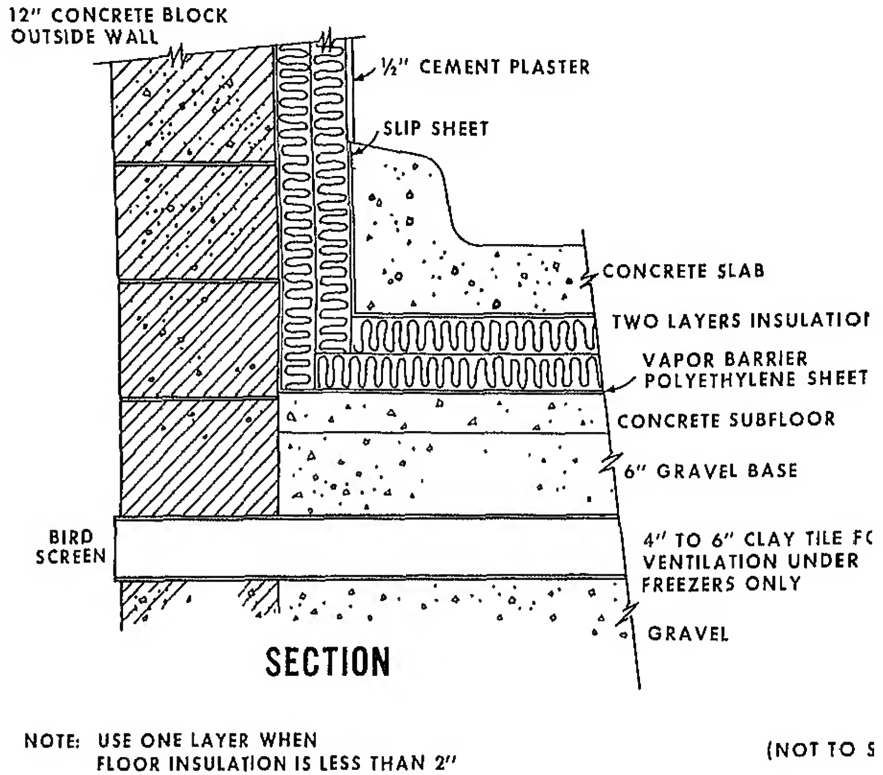

FIGURE 12.—Details of a typical floor installation

the entire year (8,760 hours), than on the outside wall, since the outside tem ture exceeds 50° only 4,629 hours per year.

The same temperature conditions exist for each situation, but a room of a c temperature using a unitary package system requires a thicker insulation that required by the same room connected to a central system. Compare the i tion used on a wall between a 40° F. room and an unconditioned space, desig by code letter "L." The results:

Unitary package system-----5.5-inch expanded polystyrene.
One central system----------3.5-inch expanded polystyrene.
Four central systems--------4.0-inch expanded polystyrene.

These differences in thicknesses can be explained by referring back to the com analysis on "optimum thickness" of insulation. Since the costs of providing refri tion are lower with central systems, it costs less to maintain the 40° F. temper by using more refrigeration than by using thicker insulation.

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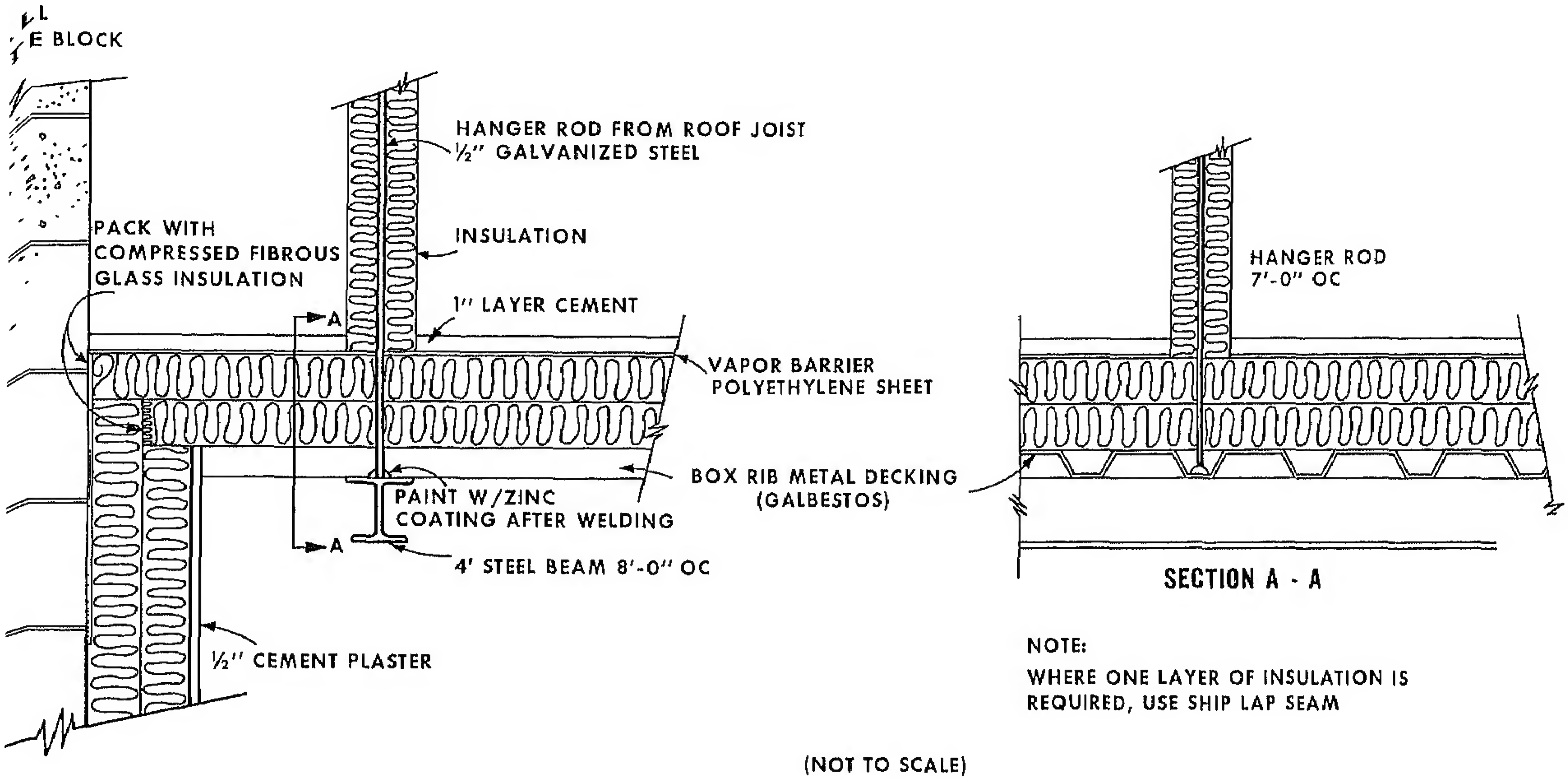


FIGURE 13.—Details of a typical freezer-ceiling installation.

TABLE 3.—*Wall-insulation schedule for Situation* I (*unitary package system of refrigeration*)

[Chicago weighted-temperature-hour approach used to determine thickness]

Code letter	Temperature range across wall (° F.)	WTD Q/A (B.t.u./hr.-ft.²)	Standard Q/A (B.t.u./hr.-ft.²)	Total installed cost ($/ft.²)	Insulation thickness and type
A	72 to outside	0.745	2.99	0.918	1.5-in. expanded polystyrene.
B	50 to outside	1.254	3.32	1.206	3.0-in. expanded polystyrene.
C	45 to outside	1.410	3.21	1.248	3.5-in. expanded polystyrene.
D	40 to outside	1.42	3.50	1.248	3.5-in. expanded polystyrene.
E	32 to outside	1.015	2.59	1.495	5.0-in. fibrous glass.
F	25 to outside	1.095	2.42	1.56	6.0-in. fibrous glass.
G	−10 to outside	1.191	1.83	1.852	10.5-in. fibrous glass.
H	−20 to outside	1.261	2.05	1.917	11.5-in. fibrous glass.
I	72 to unconditioned		.818	.961	2.0-in. expanded polystyrene.
J	50 to unconditioned		1.417	1.376	5.0-in. expanded polystyrene.
J-1	50 to 72		1.252	1.291	4.0-in. expanded polystyrene.
K	45 to unconditioned		1.645	1.376	5.0-in. expanded polystyrene.
L	40 to unconditioned		1.718	1.418	5.5-in. expanded polystyrene.
L-1	40 to 72		1.483	1.376	5.0-in. expanded polystyrene.
L-2	40 to 50		.813	1.163	2.5-in. expanded polystyrene.
M	32 to unconditioned		1.296	1.69	8.0-in. fibrous glass.
M-1	32 to 50		.799	1.462	4.5-in. fibrous glass.
N	25 to unconditioned		1.394	1.722	8.5-in. fibrous glass.
N-1	25 to 72		1.329	1.657	7.5-in. fibrous glass.
N-2	25 to 40		.728	1.43	4.0-in. fibrous glass.
N-3	25 to 50		1.215	1.43	4.0-in. fibrous glass.
O	−10 to unconditioned		1.474	2.015	13.0-in. fibrous glass.
O-1	−10 to 72		1.383	1.982	12.5-in. fibrous glass.
O-2	−10 to 50		1.160	1.852	10.5-in. fibrous glass.
O-3	−10 to 40		1.061	1.787	9.5-in. fibrous glass.
O-4	−10 to 32		.981	1.722	8.5-in. fibrous glass.
O-5	−10 to 25		.817	1.722	8.5-in. fibrous glass.
P	−20 to unconditioned		1.564	2.047	13.5-in. fibrous glass.
P-1	−20 to 72		1.48	2.015	13.0-in. fibrous glass.
P-2	−20 to 50		1.292	1.885	11.0-in. fibrous glass.
P-3	−20 to 40		1.145	1.852	10.5-in. fibrous glass.
P-4	−20 to −10				None

TABLE 4.—*Ceiling- and floor-insulation schedule for Situation* I (*unitary p*[illegible] *system of refrigeration*)

[Chicago weighted-temperature-hour approach used to determine thickness]

Room temperature (° F.)	WTD Q/A (B.t.u./hr.-ft.²)	Standard Q/A (B.t.u./hr.-ft.²)	Total installed cost ($/ft.²)	Insulation thickness and ty
Ceiling-insulation:				
72	1.962	3.82	1.323	4.5-in. expanded poly
50	2.059	4.42	1.366	5.0-in. expanded poly
45	2.102	4.07	1.408	5.5-in. expanded poly
40	2.042	4.35	1.408	5.5-in. expanded poly
32	1.354	3.14	1.657	7.5-in. fibrous glass.
25	1.456	3.12	1.69	8.0-in. fibrous glass.
−10	1.358	2.54	1.982	12.5-in. fibrous glass.
−20	1.447	2.59	2.015	13.0-in fibrous glass.
Floor-insulation:				
50		0.63	0.253	1.5-in. expanded poly
45		.845	.338	2.5-in. expanded polys
40		.95	423	3.5-in expanded polys
32		1.054	.622	5.0-in expanded polys
25		1.251	.664	5.5-in expanded polys
−10		1.49	1.047	10.0-in expanded polys
−20		1.55	1.132	11.0-in. expanded polys

TABLE 5.—*Wall-insulation schedule for Situation II (central system of refrigeration for four buildings)*

[Chicago weighted-temperature-hour approach used to determine thickness]

Code letter	Temperature range across wall (° F.)	WTD Q/A (B.t.u./hr.-ft.²)	Standard Q/A (B.t.u./hr.-ft.²)	Total installed cost ($/ft.²)	Insulation thickness and type
A	72 to outside	1.417	5 67	0.833	0.5-in. expanded polystyrene.
B	50 to outside	1.739	4.60	1.121	2.0-in. expanded polystyrene.
C	45 to outside	2.239	5.07	1.121	2.0-in. expanded polystyrene.
D	40 to outside	2.248	5.55	1.121	2.0-in. expanded polystyrene.
E	32 to outside	2.052	5.23	1.244	2.5-in. expanded polystyrene.
F	25 to outside	2.248	4.96	1.287	3.0-in. expanded polystyrene.
G	−10 to outside	2.193	3.35	1.527	5.5-in. fibrous glass.
H	−20 to outside	2 332	3.78	1.56	6.0-in. fibrous glass.
I	72 to unconditioned		1.303	.876	1.0-in. expanded polystyrene.
J	50 to unconditioned		2.19	1.206	3.0-in. expanded polystyrene.
J-1	50 to 72		1.845	1.163	2.5-in. expanded polystyrene.
K	45 to unconditioned		2.543	1.206	3.0-in. expanded polystyrene.
L	40 to unconditioned		2.544	1.248	3.5-in. expanded polystyrene.
L-1	40 to 72		2.295	1.206	3.0-in. expanded polystyrene.
L-2	40 to 50		1.194	1.078	1.5-in. expanded polystyrene.
L-3	40 to 45				None.
M	32 to unconditioned		3.029	1.329	3.5-in. expanded polystyrene.
M-1	32 to 50		1.729	1.202	2.0-in. expanded polystyrene.
N	25 to unconditioned		3.076	1.372	4.0-in. expanded polystyrene.
N-1	25 to 72		2.921	1.329	3.5-in. expanded polystyrene.
N-2	25 to 40		1.417	1.202	2.0-in. expanded polystyrene.
N-3	25 to 50		1.575	1.285	3.0-in. expanded polystyrene.
O	−10 to unconditioned		2.645	1.625	7.0-in. fibrous glass.
O-1	−10 to 72		2.561	1.592	6.5-in. fibrous glass.
O-2	−10 to 50		2.138	1.527	5.5-in. fibrous glass.
O-3	−10 to 40		1.924	1.495	5.0-in. fibrous glass.
O-4	−10 to 32		1.763	1.462	4.5-in. fibrous glass.
O-5	−10 to 25		1.47	1.462	4.5-in. fibrous glass.
P	−20 to unconditioned		2.73	1.657	7.5-in. fibrous glass.
P-1	−20 to 72		2.657	1.625	7.0-in. fibrous glass.
P-2	−20 to 50		2.281	1.56	6.0-in. fibrous glass.
P-3	−20 to 40		2.096	1.527	5.5-in. fibrous glass.
P-4	−20 to −10				None.

TABLE 6.—*Ceiling- and floor-insulation schedule for Situation II (central system of refrigeration for four buildings)*

[Chicago weighted-temperature-hour approach used to determine thickness]

Room temperature (° F.)	WTD Q/A (B.t.u./hr.-ft.²)	Standard Q/A (B.t.u./hr.-ft.²)	Total installed cost ($/ft.²)	Insulation thickness and type
Ceiling insulation:				
72	2.79	5.43	1.196	3.0-in. expanded polystyrene.
50	3.225	6.93	1.196	3.0-in. expanded polystyrene.
45	3.146	6.38	1.238	3.5-in. expanded polystyrene
40	3.058	6.63	1.238	3.5-in. expanded polystyrene.
32	3.037	7.05	1.238	3.5-in. expanded polystyrene.
25	3.083	6.60	1.362	4.0-in. expanded polystyrene.
−10	2.516	4.75	1.592	6.5-in. fibrous glass.
−20	2.616	4.68	1.625	7.0-in. fibrous glass.
Floor insulation:				
72				None.
50		0.832	0.211	1.0-in. expanded polystyrene.
45		1.255	.253	1.5-in. expanded polystyrene.
40		1.508	.296	2.0-in. expanded polystyrene.
32		1.918	.409	2.5-in. expanded polystyrene.
25		2.134	.452	3.0-in. expanded polystyrene.
−10		2.603	.664	5.5-in. expanded polystyrene.
−20		2.74	.707	6.0-in expanded polystyrene.

TABLE 7. *Wall-insulation schedule for Situation III (central system of refrigeration for each of four buildings)*

[Chicago weighted-temperature-hour approach used to determine thickness]

Code letter	Temperature range across wall (° F.)	WTD Q/A (B.t.u./hr.-ft.²)	Standard Q/A (B.t.u./hr.-ft.²)	Total installed cost ($/ft.²)	Insulation thickness and type
A	72 to outside	0.976	3.92	0.876	1.0-in. expanded polystyrene.
B	50 to outside	1.739	4.60	1.121	2.0-in. expanded polystyrene.
C	45 to outside	1.876	4.25	1.163	2.5-in. expanded polystyrene.
D	40 to outside	1.883	4.65	1.163	2.5-in. expanded polystyrene.
E	32 to outside	1.764	4.50	1.287	3.0-in. expanded polystyrene.
F	25 to outside	1.971	4.35	1.329	3.5-in. expanded polystyrene.
G	−10 to outside	1.753	2.68	1.625	7.0-in. fibrous glass.
H	−20 to outside	1.894	3.08	1.657	7.5-in. fibrous glass.
I	72 to unconditioned		1.005	.918	1.5-in. expanded polystyrene.
J	50 to unconditioned		1.927	1.248	3.5-in. expanded polystyrene.
J-1	50 to 72		1.593	1.206	3.0-in. expanded polystyrene.
K	45 to unconditioned		2.237	1.248	3.5-in. expanded polystyrene.
L	40 to unconditioned		2.271	1.291	4.0-in. expanded polystyrene.
L-1	40 to 72		2.019	1.248	3.5-in. expanded polystyrene.
L-2	40 to 50		1.194	1.078	1.5-in. expanded polystyrene.
L-3	40 to 45				None.
M	32 to unconditioned		2.441	1.414	4.5-in. expanded polystyrene.
M-1	32 to 50		1.453	1.244	2.5-in. expanded polystyrene.
N	25 to unconditioned		2.777	1.414	4.5-in. expanded polystyrene.
N-1	25 to 72		2.606	1.372	4.0-in. expanded polystyrene.
N-2	25 to 50		1.417	1.202	2.0-in. expanded polystyrene.
N-3	25 to 50		2.46	1.202	2.0-in. expanded polystyrene.
O	−10 to unconditioned		2.207	1.722	8.5-in. fibrous glass.
O-1	−10 to 72		2.112	1.69	8.0-in. fibrous glass.
O-2	−10 to 50		1.834	1.592	6.5-in. fibrous glass.
O-3	−10 to 40		1.629	1.56	6.0-in. fibrous glass.
O-4	−10 to 32		1.47	1.527	5.5-in. fibrous glass.
O-5	−10 to 25		1.225	1.527	5.5-in. fibrous glass.
P	−20 to unconditioned		2.301	1.755	9.0-in. fibrous glass.
P-1	−20 to 72		2.216	1.722	8.5-in. fibrous glass.
P-2	−20 to 50		1.978	1.625	7.0-in. fibrous glass.
P-3	−20 to 40		1.798	1.592	6.5-in. fibrous glass.
P-4	−20 to −10				None.

TABLE 8.—*Ceiling- and floor-insulation schedule for Situation III (central syste refrigeration for each of four buildings)*

[Chicago weighted-temperature-hour approach used to determine thickness]

Room temperature (° F.)	WTD Q/A (B.t.u./hr.-ft.²)	Standard Q/A (B.t.u./hr.-ft.²)	Total installed cost ($/ft.²)	Insulation thickness and type
Ceiling insulation:				
72	2.79	5.43	1.196	3.0-in. expanded polysty
50	2.513	5.40	1.281	4.0-in. expanded polysty
45	2.799	5.67	1.281	4.0-in. expanded polysty
40	2.72	5.90	1.281	4.0-in. expanded polysty
32	2.70	6.26	1.362	4.0-in. expanded polysty
25	2.775	5.95	1.404	4.5-in. expanded polysty
−10	2.068	3.90	1.69	8.0-in. fibrous glass.
−20	2.176	3.90	1.722	8.5-in. fibrous glass.
Floor insulation:				
72				None.
50		0.832	0.211	1.0-in. expanded polysty
45		1.01	.296	2.0-in. expanded polysty
40		1.261	.338	2.5-in. expanded polysty
32		1.648	.452	3.0-in. expanded polysty
25		1.87	.494	3.5-in. expanded polysty
−10		2.232	.749	6.5-in. expanded polysty
−20		2.375	.792	7.0-in. expanded polysty

DETERMINING THE REFRIGERATION LOADS

Before the refrigeration-load calculations for this study were completed, visits were made to relatively new market areas of three large eastern cities to review the problems with existing refrigeration systems and to obtain recommendations for improvements. Inadequate refrigeration and storage were evident in many instances, and modern materials-handling systems were not in full use.

Inadequate refrigeration is often the result of design, which relates back to improper calculation of the refrigeration load for which the equipment was select The total refrigeration load is based on heat gains from four general sourc (1) Transmission through walls, floors, and ceilings; (2) air changes; (3) cooli of products; and (4) miscellaneous loads from such sources as lights, motors, a

people within the refrigerated space Each type of heat gain and the parameters as used for this study are discussed in this section.

Failure to use modern materials-handling systems often reduces the effective utilization of the storage areas as well as the efficiency of handling. Unless otherwise noted, the capacity of the storage being considered is evaluated on the number of 40- × 48-inch pallets that can be accommodated. An overall face width of 57 inches or more is used so that pallet racks can be employed. Pallets are arranged two high in the 10- and 12-foot-high storage rooms, and three high in the 20-foot-high storage areas. Aisles about 8 feet wide are provided for forklift trucks.

Table 9 illustrates the type of worksheet that was set up for tabulating heat gains for the various sources as applied to Situations I, II, and III. This form represents a worksheet that can be used in calculating the refrigeration load for each refrigerated space within the distribution complex, but it refers specifically to Building No. 1, Firm No. 4, Room No. 1. All sample calculations that follow refer to this table.

Heat Gains from Transmission Through Walls, Floor, and Ceiling

Any heat gains through the walls, floor, and ceiling will vary with the type and thickness of construction materials, including insulation; with the surface area and with the temperature difference between the refrigerated space and the air on the other side of the surface. All of these factors are expressed in the formula

$$Q = U \times A \times \Delta t^{7}$$

where:

Q = heat gain in B.t.u./hr.
U = coefficient of heat transmission, in B.t.u./hr.-ft.^{2}-° F
A = outside area of the surface in ft.^{2}
Δt = difference between two air temperatures

[7] ASHRAE Handbook of Fundamentals, 1967, ch. 29, p. 513.

TABLE 9.—*Sample refrigeration load worksheet—Building No. 1, Firm No. 4, Room No. 1, (24′ × 70′ × 20′), (32° F., 85 percent RH)*

	Area (ft.^{2})	Insulation code	Situation I: Std. Q/A (B.t.u./hr.-ft.^{2})	Situation I: Q (B.t.u./hr.)	Situation II: Std. Q/A (B.t.u./hr.-ft.^{2})	Situation II: Q (B.t.u./hr.)	Situation III: Std. Q/A (B.t.u./hr.-ft.^{2})	Situation III: Q (B.t.u./hr.)
Transmission:								
Walls:								
North	480	E	2.59	1,245	5.23	2,510	4.50	2,160
South	480	E	2.59	1,245	5.23	2,510	4.50	2,160
East	1,400	M	1.296	1,810	3.029	4,240	2.441	3,420
West	1,400	M	1.296	1,810	3.029	4,240	2.441	3,420
Roof/ceiling	1,680	32° F.	3.14	5,270	7.05	11,850	6.26	10,500
Floor	1,680	32° F.	1.054	1,770	1.918	3,220	1.648	2,770
Miscellaneous:								
Lights	1,680 w × 3.4 B.t.u./hr./w.			5,720		5,720		5,720
Motors	4.5 hp. × 4,150 B.t.u./hr./hp.			18,700		18,700		18,700
People	2 × 800 B.t.u./hr./person			1,600		1,600		1,600
Air changes:	No./hr. × room volume × B.t.u./ft.^{3}							
	0.212 × 33,600 ft.^{3} × 2.51			17,880		17,880		17,880
Product load:								
No. of pallets	216							
Weight/pallet	1,600 lb.							
Total product weight	345,600 lb.							
Turnover	5 days							
Cooling range	30° F.							
Specific heat above freezing	0.9 B.t.u./lb.-° F.			77,700		77,700		77,700
Respiration	3,500 B.t.u./ton/day			25,180		25,180		25,180
Latent heat of fusion	B.t.u./lb.			----------		----------		----------
Specific heat below freezing	B.t.u./lb.-° F.			----------		----------		----------
Freezing capacity	lb./hr.			----------		----------		----------
Total refrigeration load B.t.u./hr.				159,850		175,350		171,310

[illegible] represented by (U × Δt) has already been considered in the [illegible] tables 3, 4, 5, 6, 7, and 8, where it is listed as Standard Q/A (B.t.u./[illegible]).

[illegible] an attempt was made to eliminate all variable data that [illegible] the final solution. Computer runs indicated that factors such as [illegible] of the outer walls, sun heat loads on the vertical [illegible] total effect that their elimination [illegible] alter the result. The effect of the sun on the roof was appreci[illegible] included.

[illegible] on the basis of an outside [illegible] temperature of 95° F. dry bulb. See figure 11A for the construction of an [illegible] in this study.

[illegible] on the basis of an outside design [illegible] dry bulb with [illegible] penalty for solar load. See figure 13 for [illegible] in the study.

[illegible] on a constant temperature [illegible] are assumed to have a [illegible] F. dry bulb.

[illegible] a constant underfloor tempera[illegible] F. dry bulb [illegible] assumed for this [illegible].

[illegible] the walls, floor, and ceiling are [illegible]

[illegible]

[illegible] from table 4, [illegible]

[illegible]

[illegible] Situations II and III, the wall [illegible] Std Q/A [illegible] listed in tables 5 and 7.

[illegible] Situations I, II, and III, because the [illegible] systems in II and III affect [illegible] insulation required allows greater [illegible]

[illegible] used for Situations II and III [illegible] over that used in Situation I, the heat gains [illegible] percent for rooms above 32° and 2 to 4 percent for freezers. [illegible] at an outside design temperature of 95° F., [illegible] for a few hours each year in the Chicago area.

The total heat gain from transmission through the walls, floor, and ceiling [illegible] of the overall refrigeration load.

Heat Gains from Air Changes

Each time that the door to a refrigerated room is opened, some w[illegible] air enters the room. This air must be cooled to the room temperature, [illegible] to the refrigeration load.

The heat gain from infiltration and air changes is calculated from [illegible] and figures in the ASHRAE Handbook of Fundamentals, 1967, chapt[illegible] 513, 514, and 515.

Q (B.t.u./hr.) = No. of changes/hr. × room volume (ft.^{3}) × heat removed i[illegible] air to storage temperature (B.t.u./ft.^{3})

In the following sample calculation, the traffic into the refrigera[illegible] considered as being heavy.

$$Q = 0.212 \text{ changes/hr.} \times 33{,}600 \text{ ft.}^3 \times 2.51 \text{ B.t.u./ft.}^3$$
$$= 17{,}880 \text{ B.t.u./hr.}$$

Heat Gains from Cooling the Product

When a product is placed in a refrigerated space at a temperature [illegible] the room temperature, that product will lose heat until its temperatu[illegible] with the room temperature. The quantity of heat lost by a product dep[illegible] entering and leaving conditions, weight, specific heat above and belo[illegible] freezing point, and the amount of latent heat to be removed.

The following formulas[8] are used to calculate the total heat gain wit[illegible] where stored products are losing heat:

a. Heat gained by lowering the temperature of the entering product to some [illegible] freezing or to freezing:

$$Q = W \times c(t_1 - t_2, \text{ or } t_f)$$

b. Heat gained by freezing product:

$$Q = W \times h_{if}$$

c. Heat gained by lowering product temperature from the freezing temperature [illegible] storage temperature:

$$Q = W \times c_i\,(t_f - t_3)$$

where:

Q = heat gain (B.t.u.)
W = weight of product (lb.)
c = specific heat of product above freezing (B.t.u./lb.-° F.)
t_1 = initial entering temperature (° F.)
t_2 = final storage temperature above freezing (° F.)
t_f = freezing temperature of product (° F.)
h_{if} = latent heat of fusion (B.t.u./lb.)
c_i = specific heat of product below freezing (B.t.u./lb.-° F.)
t_3 = final storage temperature below freezing (° F.)

---

[8] ASHRAE Handbook of Fundamentals, 1967, ch. 29, p. 514.

The total product-cooling load in B.t.u.'s is represented by the sum of these individual cooling steps. Respiration-heat gains must also be included for fruits and vegetables.

Some data assumptions were made to determine the product loads for each of the buildings. (For each refrigerated space, the product load is considered to be the same, regardless of the refrigeration system.) The weight and specific heat of pallets and containers were disregarded in this study. A sample calculation follows:

$$\begin{aligned}\text{Product weight (W)} &= \text{No. of pallet loads} \times \text{weight/pallet load (lb.)}\\ &= 216 \text{ pallet loads} \times 1{,}600 \text{ lb}\\ &= 345{,}600 \text{ lb.}\\ Q \text{ (B t u./hr.)} &= W \times c\ (t_1 - t_2) \div \text{turnover (hr.)}\\ &= \frac{345{,}600 \text{ lb.} \times 0.9 \text{ B.t u /lb.-° F. } (62° - 32°)}{5 \text{ days} \times 24 \text{ hr./day}}\\ &= 77{,}700 \text{ B.t u./hr.}\end{aligned}$$

Heat gain from respiration:

$$\begin{aligned}Q \text{ (B.t.u./hr)} &= \text{respiration } \frac{\text{(B.t.u./ton/day)} \times \text{product weight (lb.)}}{24 \text{ hr./day} \times 2{,}000 \text{ lb /ton}}\\ &= \frac{3{,}500 \text{ B t.u./ton/day} \times 345{,}600 \text{ lb.}}{24 \text{ hr./day} \times 2{,}000 \text{ lb./ton}}\\ &= 25{,}180 \text{ B.t.u /hr.}\end{aligned}$$

The total product heat load can be found by adding (1) heat gain that occurs in lowering the product temperature from 62° to 32° F. to (2) the heat gain from product respiration.

Fruits and vegetables—Buildings No. 1 and No. 4

A complete product turnover every 5 days is assumed in Building No. 1 and every 4 days in building No. 4. Entering product temperature—

30° F. above the storage temperature for 32° rooms, and above.
20° F. above the storage temperature in freezers.[9]

The figures used for specific heat and heat of respiration are based on an average for the fruits and vegetables normally stored at the respective room temperatures.

Room temperature	*Specific heat (B.t.u./lb.–°F.)*	*Heat of respiration (B.t.u./ton/day)*
−10° F.	0.45	----------------
32° F.	.9	3,500
40° and 45° F.	.94	5,500
50° F.	.92	7,500

Respiration is the process by which oxygen of the air is combined with the carbon of the plant tissue, thus releasing energy in the form of heat. This heat gain must be removed by the refrigeration equipment. The colder the product, the lower its metabolic rate, and the less the heat of respiration given off.

A product weight of 1,600 pounds per pallet was assumed for all storage areas.

[9] These temperatures are not to be interpreted as recommended entering temperature values for fruits and vegetables. They are assumed values and are used solely for calculating the refrigeration loads.

Meats and meat products—Building No. 2

A complete product turnover every 3 days is assumed for all the rooms except the freezers, where a turnover every 5 days is assumed.

Entering temperature—

10° F. above the storage temperature for all rooms except freezers.
42° F. body temperature for all products entering the freezers.

Meat rails with an average product loading of 120 lb./ft. of rail are used in the various 32° F. holding rooms. A pallet weight of 1,600 pounds was used for all meat storage areas. The freezers are assumed to be for specialty freezing, with the lb./hr. freezing rates as shown on the drawing for each situation.

$$\text{Freezing rate (lb./hr.)} = \frac{\text{weight of product (lb.)}}{\text{turnover time (hr.)}}$$

The specific heat and latent heat values are averages for the various meats that would be handled:

Specific heat above freezing for—	
Carcass meats	0.77 B.t.u./lb.-° F.
Package meats	.72 B.t.u./lb.-° F.
Specific heat below freezing for all meats	.40 B.t.u./lb.-° F.
Latent heat	100 B.t.u./lb.

Poultry and eggs—Building No. 3

These spaces are laid out for pallet operation, with the exception of egg storages, which are designed for a stacking height of seven cases. Space is included to palletize egg storages if desirable. Pallet sizes of 36 × 48 inches are used for eggs and poultry, with 8 inches allowed between pallets for pallet racks and air circulation.

Freezer loads are based on a 5-day turnover, while all other products and spaces are based on a 3-day turnover.

The products enter the various spaces at the following temperatures:

Room temperature	*Entering product temperature*
−10° and 20° F.	40° F.
40° F.	50° F.
50° F.	72° F.
72° F.	82° F.

Exceptions to the above are the −10° F. freezers in Firms No. 25 and No. 26, where the product enters at 10°; and the −20° egg freezer of Firm No. 24, where the product enters at 57°.

In the 25° F. storage room of Firm No. 27, which is used for crusted chicken, a 5° temperature difference between entering product and room is assumed

Specific heat above freezing for—	
Poultry	0.79 B.t.u./lb -° F.
Eggs	.76 B.t.u./lb -° F.
Specific heat below freezing for—	
Poultry	0.37 B.t.u./lb -° F
Eggs	.41 B.t.u /lb.-° F
Latent heat of fusion—	
Poultry	106 B.t.u /lb.
Eggs	100 B.t.u /lb.
Average freezing point—	
Poultry	27° F.
Eggs	27° F.

Pallet weights of 1,600 pounds were assumed for poultry and 1,200 pounds for shell eggs.

Heat Gains from Miscellaneous Sources

Miscellaneous refrigeration loads result from heat gains produced by the electrical energy dissipated within the refrigerated space by lights, motors, heaters, etc.; and heat given off by people who enter or work within the space. These heat gains are calculated in accordance with the ASHRAE Handbook of Fundamentals, 1967, chapter 29, page 515.

A lighting load of 1 watt/square foot of floor area is used in all spaces except the cutting rooms and work areas, where 2 watts/square foot is used.

For the work areas, an additional refrigeration load is calculated for 5 horsepower of miscellaneous motors for each 1,000 square feet of floor area. One person for each 500 square feet of floor area is assumed in all cutting rooms and work areas.

The Total Refrigeration Loads by Building and Situation

:he sum of these individual heat gains is the total refrigeration load for each resentative firm. The refrigeration equipment is selected to handle this total 1.

Table 10 lists the calculated refrigeration loads by building and situ Since not all firms have peak loads simultaneously, in actual practice it is po to develop a diversity factor from central systems. A diversity factor is the ence between the maximum calculated tonnage and the maximum tonnage ac required at any one time. In this study the central systems are selected to the total of the maximum calculated loads and, therefore, have the diversity as a safety factor. Any exceptions are noted in the equipment selection s for each situation.

The refrigeration loads for each firm are listed on the individual building plans in each situation. High-stage refrigeration systems handle the refrige loads for all rooms 32° F. and above, and low-stage refrigeration systems hand refrigeration loads for all rooms under 32°.

TABLE 10.—*Summary of refrigeration loads by building and situation*

Building No.	Product load TR[1]	Situation I TR	Situation II TR	Situation TR
1—Total	*100*	*162.4*	*169.2*	*167.9*
High stage	100	162.4	169.2	167.9
Low stage	---	---	---	---
2—Total	*46.3*	*122.7*	*138.6*	*133.0*
High stage	12.4	80.3	92.6	88.6
Low stage	33.9	42 4	46 0	44.4
3—Total	*51.6*	*108 8*	*118 1*	*114.6*
High state	22.9	62 0	68.4	66.0
Low stage	28.7	46 8	49.7	48.6
4—Total	*18.4*	*32 9*	*36 0*	*34.9*
High stage	16.7	26.1	28.4	27.7
Low stage	1.7	6.8	7.6	7.2
Total TR		426.8	461.9	450.4
Total high-stage TR[2]		330.8	358.6	350.2
Total low-stage TR[3]		96.0	103.3	100.2

[1] TR—Tons of refrigeration

[2] High-stage refrigeration loads include all rooms 32° F and above.

[3] Low-stage refrigeration loads are for rooms under 32° F

SELECTING A REFRIGERANT

This part of the study evaluates the commonly used commercial refrigerants nd selects the type or types to be used in each of the three situations under conideration.

A number of fluids have properties that make them suitable for use as refrigerants. When all the characteristics of a given refrigeration system are considered, however, usually only very few refrigerants are desirable for the particular applicatio sometimes only one. The fluorocarbons (R-12, R-22, and R-502) and ammoni (R-717) are the most commonly used refrigerants for cold-storage applications.

Among the factors considered are the efficiency and economy of operation of th system that would result from the use of each of the common refrigerants. To mak

this analysis, it is customary to calculate their ideal performance under the standard conditions of 5° F. evaporator temperature and 86° condenser temperature.

Under standard conditions, there is little difference in horsepower per ton required by the refrigerants for lower tonnages. Their differences do become a major consideration on higher tonnages, however, where any increase in horsepower greatly affects the operating cost. For low tonnages, then, refrigerants are selected for other more important considerations besides efficiency and economy of operation of the system. These considerations are those properties of the refrigerant that reduce the needed size, weight, and initial cost of the refrigerating equipment, and that permit operation with a minimum of maintenance.

The Fluorocarbon Refrigerants

All unitary package systems are designed for use with one of the fluorocarbon refrigerants, either R–12, R–22, or R–502. It is difficult to predict which of these is the most economical, because the exact capacity and temperature level desired are not the only influencing factors. For instance, R–12, due to its stability and to its low discharge pressure with the resultant low discharge temperature from the compressor, is a favorite with owners and manufacturers of package condensing units, especially for room temperatures above 25° F. See table 11 for a comparison of suction and discharge pressures for a typical 32° room.

Table 11.—*Comparison of fluorocarbon refrigerant suction and discharge pressures*

Refrigerant	Evaporator temperature	Suction pressure	Condenser temperature	Discharge pressure
	°F.	*P.s.i.g.*[1]	*°F*	*P.s.i.g.*
R–12	20	21.0	105	125.9
R–22	20	43.0	105	210.1
R–502	20	52.5	105	229.2

[1] Pounds per square inch gage.

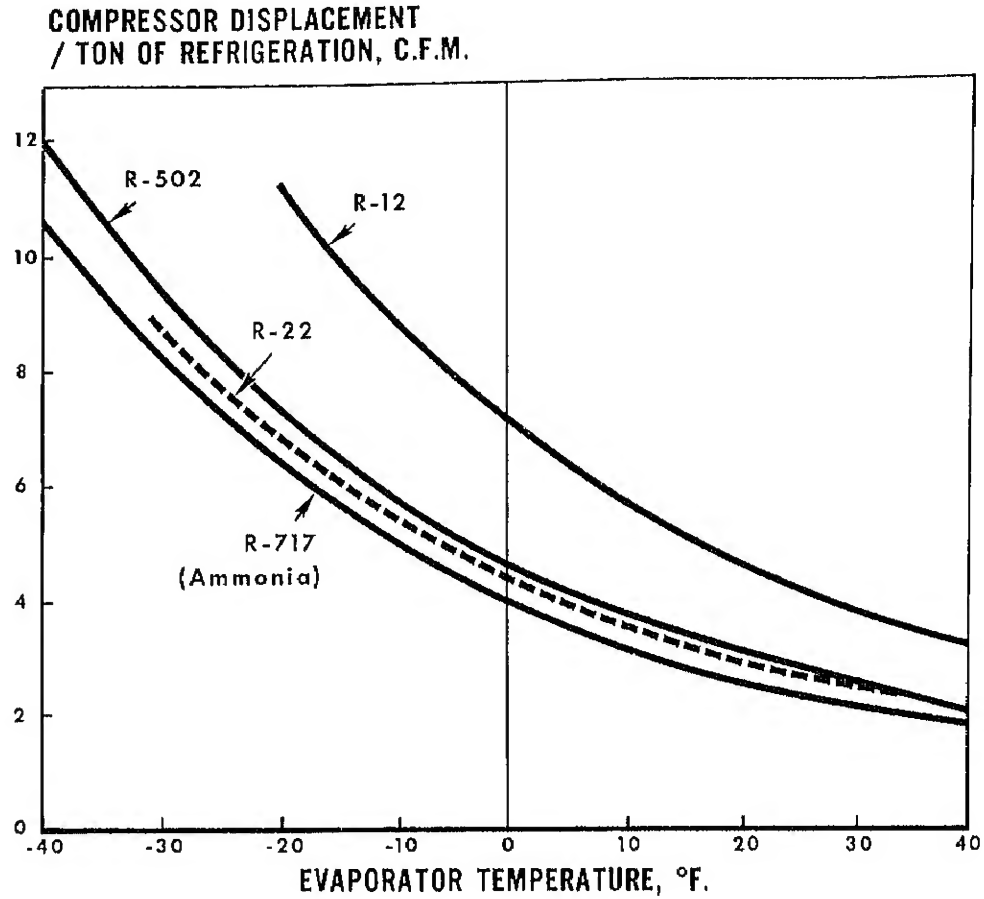


Figure 14.—Compressor volume displacement per ton of refrigeration for different refrigerants.

The above statements do not mean that R–12 is best for all higher temperature rooms. At a +20° F. evaporator temperature, a compressor must displace 4.65 c.f.m. of R–12, 2.9 c.f.m. of R–22, and 3.1 c.f.m. of R–502 gas per ton of refrigeration (fig. 14). Since the R–12 compressor must handle more gas per ton of refrigeration, its physical size must be larger for the same total tonnage. With refrigeration requirements as high as 10 to 20 tons, R-22 equipment requires less physical space than R–12 equipment.

Manufacturers tend to use the higher density[10] refrigerants, R–22 and R–502, in order to reduce compressor costs. At the lower evaporator temperatures, R–22 and R–502 are used more frequently than R–12, since it is advantageous to have the system operating above atmospheric pressure. When the system is operating in a vacuum, such as an R–12 system would be for a −20° F. room, any leak would cause atmospheric air to flow inward, which could contaminate the entire system. The repairs to a contaminated system can be quite expensive in comparison with the cost of replacing refrigerant which leaks out of a positive-pressure system, such as R–22 or R–502.

Figure 15 shows very little difference in the brake horsepower per ton of refrigeration. At the low tonnage requirements, the difference in operating costs between R–12 and R–22 is very small, but it does become significant as the tonnage increases.

[10] Density in lb./ft.^{3} is the reciprocal of the specific volume in ft.^{3}/lb.

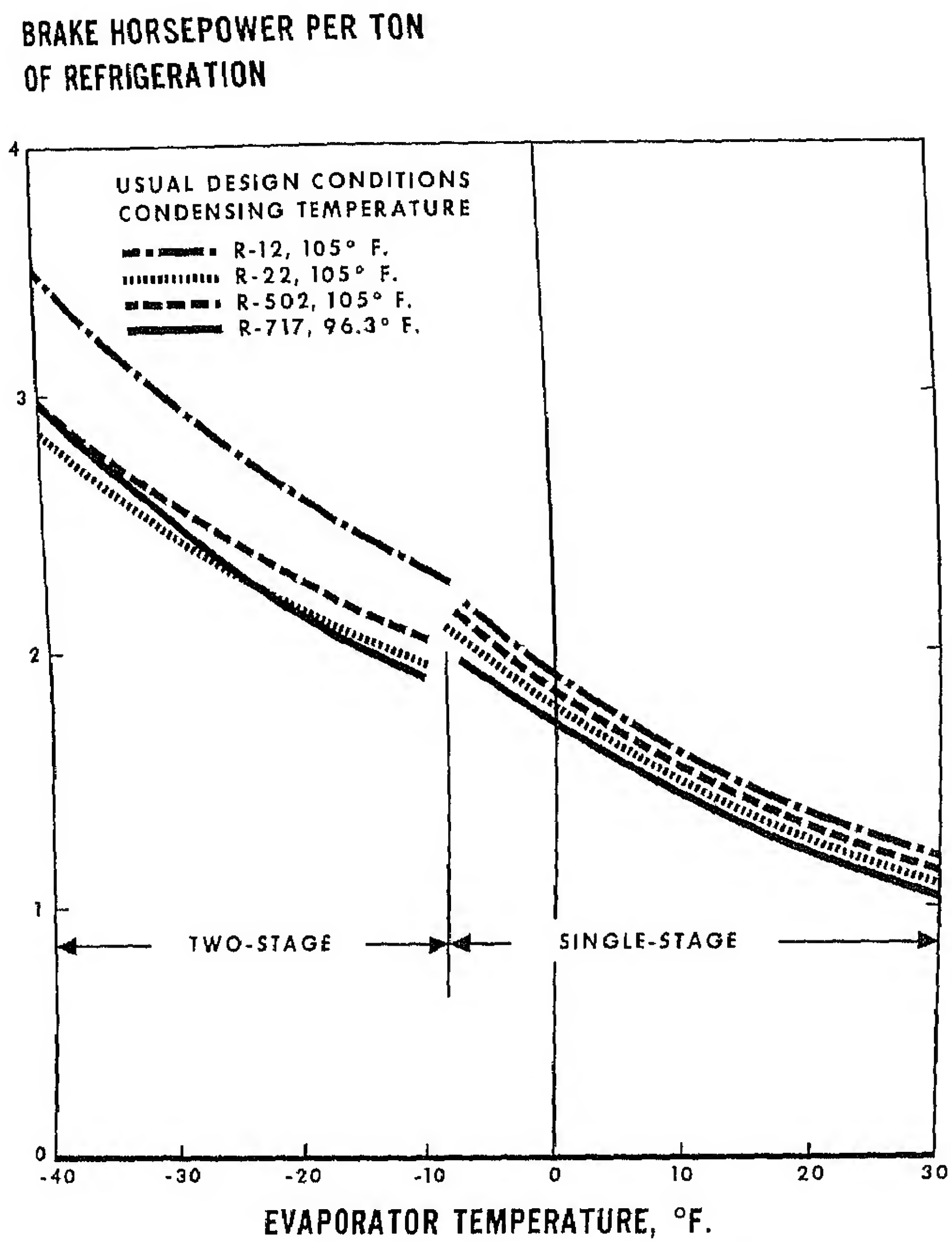


FIGURE 15.—Brake horsepower required per ton of refrigeration for different refrigerants in single- and double-stage operation.

Ammonia as a Refrigerant

Ammonia, R–717, has the lowest c.f.m./TR requirements of the four refı considered (fig. 14). In addition, its brake horsepower/ton of refrigeration TR) is the lowest of the various refrigerants, except at evaporator temp below −25° F., where R–22 has some advantages (fig. 15). In central or systems, the refrigeration tonnages become quite large, so any saving in b between different refrigerants produces significant savings in operating co lower volume of refrigerant gas circulated also results in savings on equipme

An ammonia system offers additional advantages through simplicity of installation, and maintenance. Positive oil return in a fluorocarbon syste presents some difficulties. Refrigerant–717 does not mix with the con lubricating oil, and any entrained oil separates as soon as its temperature is l An oil-drain valve is usually provided on R–717 systems at the low point evaporator and at any low points in the interconnecting lines where flow r low.

Single-stage vs. Compound Compression

Many manufacturers of package refrigeration equipment use single-stag for evaporator temperatures as low as −30° or −40° F. Use of these units is ge not considered good operating practice, considering the high compression the resultant high discharge temperatures, and the decreased reliability as ε of operating near the limits of the compressor. Low-temperature, singl compressors have higher operating costs than two-stage systems (fig. 15). lines on figure 15 for the single-stage equipment were extended to lower evaŗ temperatures, the b hp./TR would increase, which means that the operatin would go up.

Figure 15 shows that at evaporator temperatures below 0° F., compound pression equipment operates at a lower b.hp./TR. There has been hesitancy compound compression equipment on low-tonnage, low-temperature applic because of the increased costs in supplying two compressors, intercoolers, addi piping, wiring, etc. One solution to the problem is to use a single compı internally compounded. This type uses some of its cylinders to compress tl from a low evaporator temperature such as −40° up to 0°. The other cyli then compress the gas from the interstage level of 0° up to the condensing tem ture, which is approximately 105° for air-cooled condensing units. The stage suction temperature, the interstage temperature, and the condensing tem ture vary with each application, but all compression occurs within a single ur

Refrigerants Used for Package Systems, Situation I

In Situation I, it is necessary to use one of the fluorocarbon refrigerants, bec all unitary package equipment is designed for their use.

R–12 is used primarily for all rooms of 32° F. and above unless a large cooling load exists, in which case R–22 is used because of the smaller equipment required and the savings on brake horsepower (fig. 14 and 15). R–22 is used for all rooms colder than 32°, except when semihermetic condensing units are required. Here, R–502 is occasionally used for lower temperatures because of its increased cooling capacities and lower discharge temperatures. R–502 is a relatively new refrigerant on the market.

In the unitary package system (Situation I) the final selection of the refrigerant depends on the particular conditions and the sizes of equipment available.

Refrigerant Used for Central Systems in Situations II and III

Ammonia, R–717, was selected as the most economical and efficient refrigerant for use in central systems as described for Situations II and III.

The most desirable distribution system, considering oil removal and maintenance, is the pump-feed liquid-ammonia recirculation system. This system is chosen over the systems using a secondary refrigerant because of the lower first costs and operating costs.

In the pump-feed liquid-ammonia recirculation system, the oil-rich refrigerant is brought from the receiver to the low-pressure pump accumulator (via the liquid intercooler in compound systems), where it is cooled to the evaporator temperature. At this point the oil drops out and proceeds to the low point of the accumulator. If the accumulator is properly designed, only the refrigerant is pumped to the evaporator coils, eliminating the necessity for providing drain valves at the evapora tors. Drain valves may be required in designs where electric or water defrost i used. Where hot-gas defrosting is used, as in this study, it is good practice to provid a relief regulator from the low point of the evaporator to return any oil that ma have accumulated in the evaporator during the defrost cycle. These regulators ar usually set at 80 to 90 pounds per square inch gage (p.s.i.g.) on ammonia systems As the hot gas increases the evaporator pressure, the warm liquid refrigerant and oi mixture is bled into the suction lines and returned to the pump accumulator.

Oil is not easily removed from the accumulator or evaporators at temperature below −20° F. and correspondingly low operating pressures. Oil stills are recom mended to insure proper oil return on all low-temperature applications.

In some metropolitan areas, local safety codes require that an operator be avail able continuously for systems using ammonia. A licensed operator qualified t provide routine maintenance and minor repairs is considered necessary for th central systems, Situations II and III. It is also considered necessary to provide semiskilled mechanic in attendance during operation, even though the syster might be designed for automatic operation. The need for such attendants woul hold true for any large system, regardless of refrigerants and codes. The salarie for these men are included in the overall costs for central systems.

The cost of maintenance, based on what an outside firm would charge for a con tract that includes all labor, materials, and routine preventive maintenance, coul be used by the food distribution center to hire and maintain its own maintenanc engineering staff.

SITUATION I, UNITARY PACKAGE SYSTEMS

There are 34 firms located within the four-building complex of this food distribution center. Thirty-one have refrigeration requirements, which are met by one or more individual package systems for each refrigerated room (fig. 16).

A unitary package system consists of two pieces of equipment—an air-handling unit and a condensing unit. The air-handling unit is mounted in the refrigerated space for the purpose of cooling and circulating the air. It consists of a fan(s), a fan motor(s), a thermal expansion valve(s), a hand valve(s), a thermostat, and controls. The air-cooled condensing unit is located in a room in the utility tunnel beneath the rear loading platform (fig. 17). It consists of compressor, air-cooled condenser, receiver, valves, angle drier, indicator, three-valve bypass for the drier, crankcase regulator for rooms 32° F. and lower, crankcase heater and relay, starter, pressure stabilizer, and check valves for the winter head-pressure control.

Air conditioning and heating for office areas are not included for any firms in Situation I. When package units are used for refrigeration, separate units, at additional cost, must be installed for air-conditioning duty.

Equipment Selection and Operation

The air-handling equipment is selected to handle the cooling-load and ai circulation requirements. All units are selected on a 7° to 10° F. temperature di ferential, except for freezers, 72° rooms, and dry 50° rooms, where a temperatu differential of 15° or less is used. The temperature differential is the room temper ture minus the liquid-refrigerant temperature within the evaporator coil. Selectior are made for peak summer conditions and maximum product loading to make su that humidities of 85 percent and 90 percent can be maintained where require Separate humidification equipment is not necessary for short-term storage, sin packaged refrigeration equipment is capable of maintaining the high humidit required when selected with a proper temperature differential.

All air units operating in rooms 32° F. and lower use hot gas from the compress for defrosting the evaporator coil (fig. 18). An average of 2 hours per day is enoug to defrost the coils. Where the temperature is above 40°, defrosting is not necessar

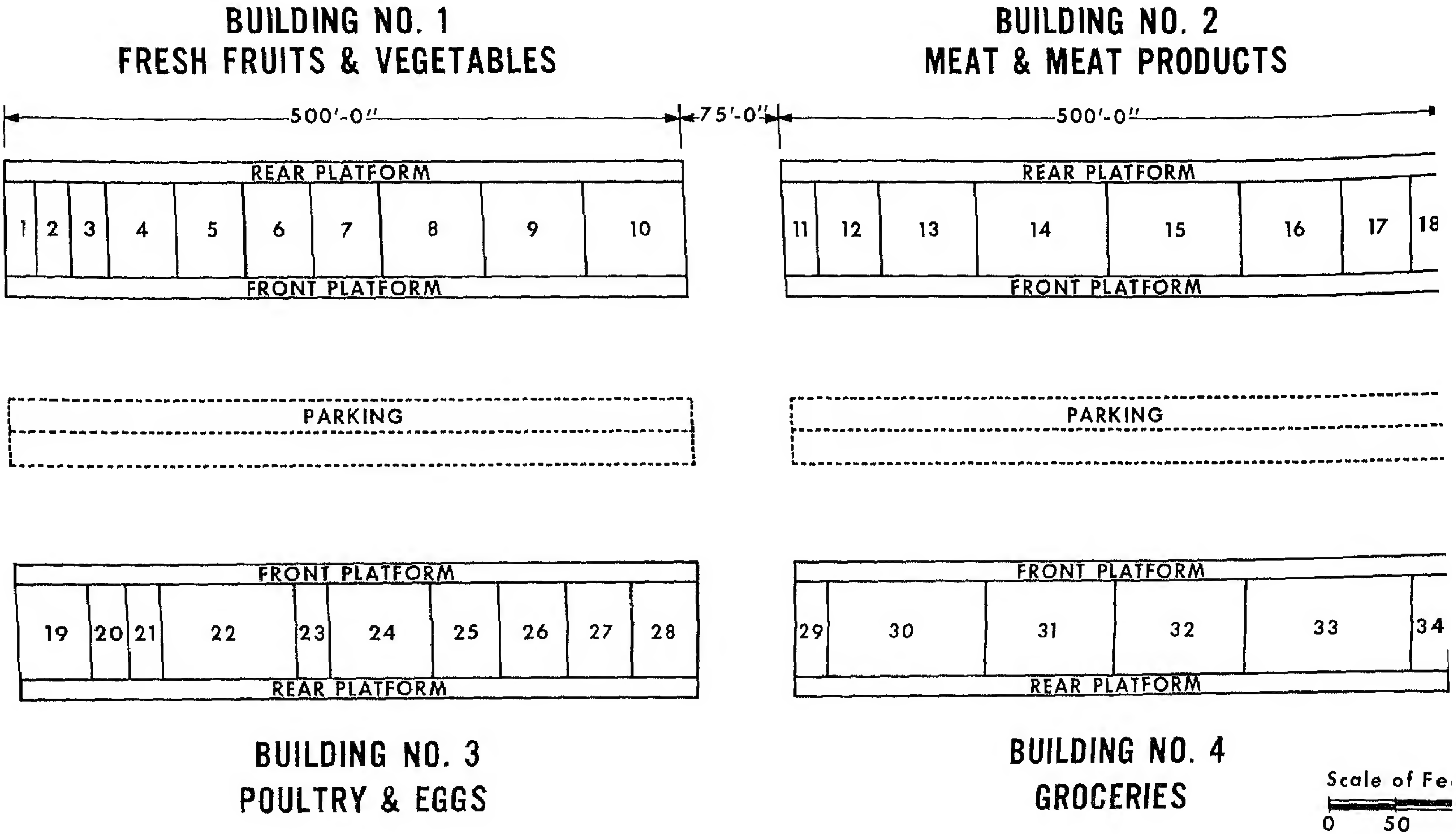


FIGURE 16.—Situation I, plot plan of food distribution center.

The air-cooled condensing units are selected in multiples per system so that up to 60 percent capacity can be maintained if one unit is not operating. This safety factor is usually adequate.

Wherever feasible, the condensing units employ open-type, direct-drive, reciprocating compressors with standard, open drip-proof motors, thereby avoiding the complications that result from semihermetic compressor motor burnouts. The open-type condensing units cost about 10 percent more than semihermetic equipment. Should the motor burn out, however, it is easier, quicker, and less exp to replace the motor of an open-style unit than it is to remove the complet pressor assembly on the semihermetic unit, send it out for repairs, and ther out the refrigerant circuit to remove any foreign matter before restarting system flushout is not necessary after a motor failure of the open-type syster

For rooms 32° F. and above, package condensing units, used in multiple usually more economical than small central or built-up systems if the total to

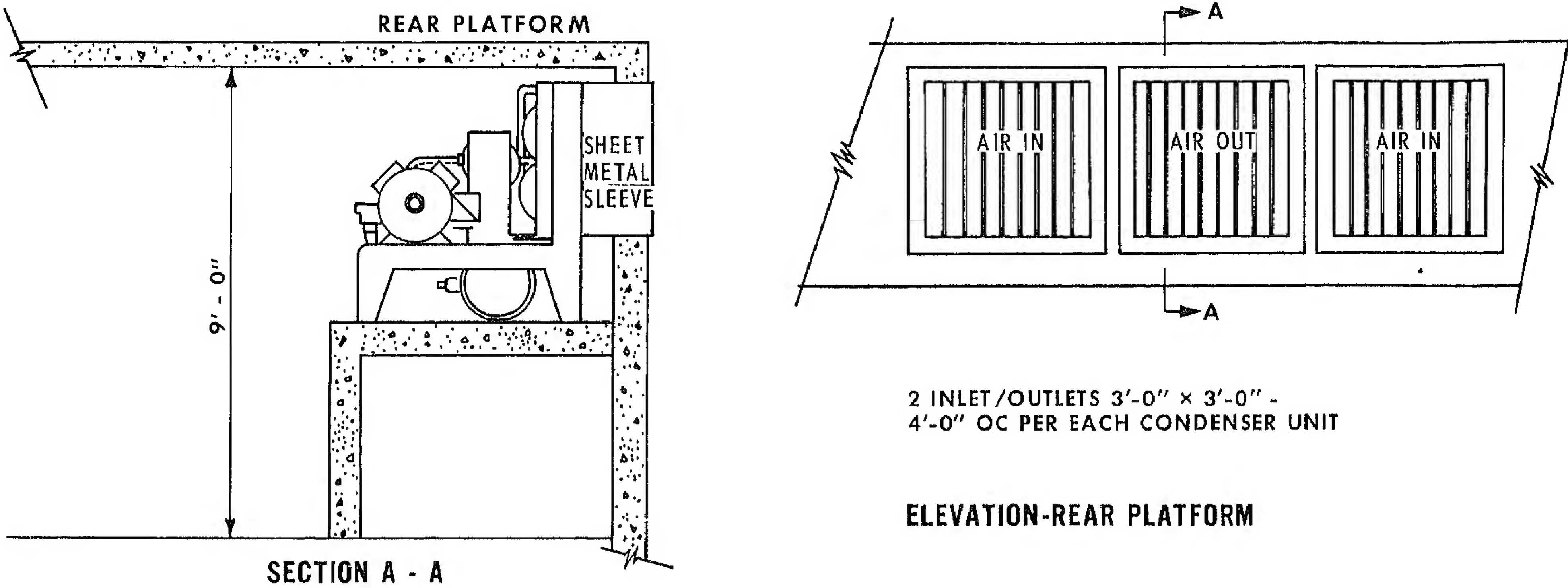


FIGURE 17.—Typical condensing unit installation.

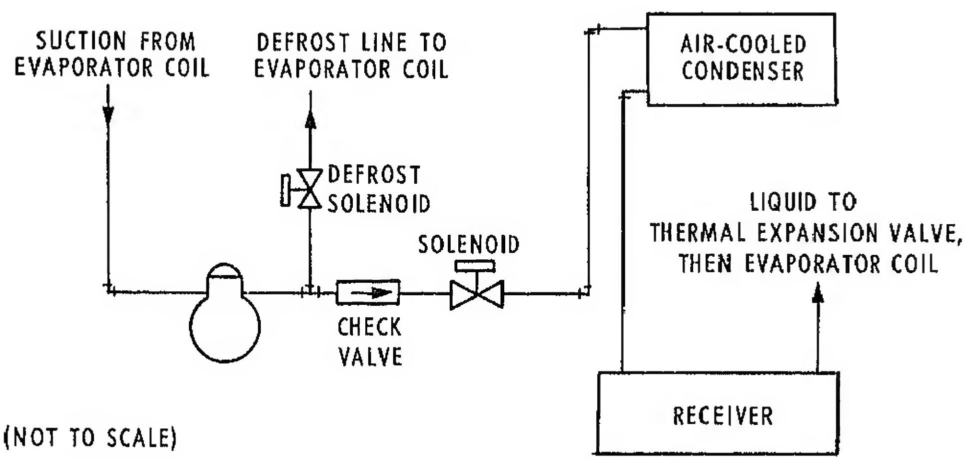


FIGURE 18.—Hot-gas defrost arrangement for rooms 32° F.

does not exceed 30 to 40 tons. Between 30 to 40 tons, a close review is needed to determine the most economical system. Above this range, the built-up system serves the purpose better. A careful review is also required for freezers with refrigeration loads up to 15 tons. For higher tonnages in this low-temperature range a built-up system is usually more economical and flexible, while lower tonnage applications can use either a package or built-up system.

Situation I specifies that multiple condensing units be used for each room which eliminates small central or built-up systems.

For applications with small cooling loads, two rooms at different temperatures can be handled by one condensing unit with the use of a back-pressure regulator on the air unit of the higher temperature room. The use of a back-pressure regulator is feasible if the higher temperature room load is less than 25 percent of the lower temperature load and the load falls within the range of available condensing unit capacities. Operating cost penalties, however, may soon be enough to pay for separate condensing units to handle each room.

Floor Plans and Refrigeration Equipment Layouts

Figures 19, 20, 21, and 22 illustrate the floor plans and refrigeration equipment layouts for all firms in Buildings Nos. 1, 2, 3, and 4, respectively. Each figure

(Text continued on page 36.)

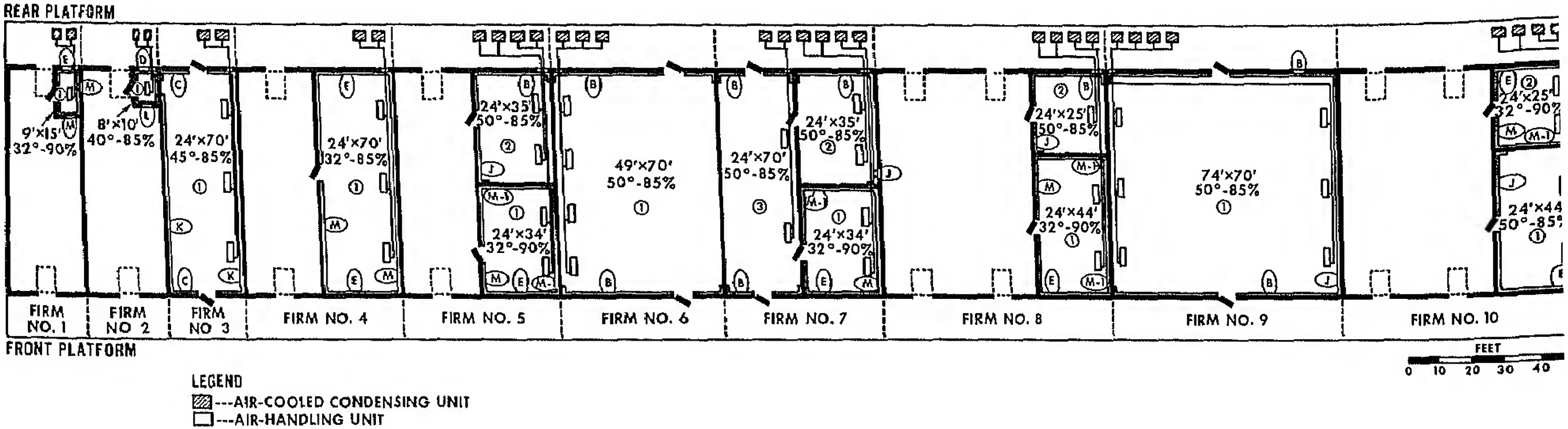


FIGURE 19.—Situation I, Building No. 1 (fresh fruits and vegetables), floor plan and refrigeration equipment location.

REFRIGERATION AND EQUIPMENT SCHEDULE

Item	Firm 1	Firm 2	Firm 3	Firm 4	Firm 5		Firm 6
	Room 1	Room 1	Room 1	Room 1	Room 1	Room 2	Room 1
Refrigeration							
Refrigeration load____B.t.u./hr._____	16,303	10,008	152,325	169,650	87,621	100,344	296,940
Equipment							
Air-handling unit:							
Model No________________________	AH-4	AH-4	AH-7	AH-10	AH-10	AH-7	AH-7
Rating____________B.t u./° F.____	1,660	1,660	6,000	6,420	6,420	6,000	6,000
Air volume______________c f m.____	3,000	3,000	10,500	12,500	12,500	10,500	10,500
Fan motor:							
No. required__________________	2	2	3	3	3	3	3
Size____________________hp____	1/12	1/12	1/2	1/2	1/2	1/2	1/2
Type of defrost__________________	Air	Air	Air	Hot gas	Hot gas	Air	Air
No. of units required_______________	1	1	3	3	2	2	4
Condensing Unit:							
Model No________________________	CU-2-12	CU-3/4-12	CU-15-12	CU-15-22	CU-7½-12	CU-10-12	CU-15-12
No. of units required_____________	2	2	2	2	2	2	3
Total operating kw______________	1 5	1.6	18 3	23 2	13 1	12 4	36 8

Item	Firm 7			Firm 8		Firm 9	Firm 10	
	Room 1	Room 2	Room 3	Room 1	Room 2	Room 1	Room 1	Ro
Refrigeration								
Refrigeration load____B.t.u./hr._____	87,621	100,344	134,758	102,251	70,235	461,870	103,374	64,
Equipment								
Air-handling unit:								
Model No________________________	AH-10	AH-7	AH-7	AH-10	AH-7	AH-7	AH-7	A
Rating____________B.t u./° F.____	6,420	6,000	6,000	6,420	6,000	6,000	6,000	3,
Air volume______________c f m.____	12,500	10,500	10,500	12,500	10,500	10,500	10,500	7,
Fan motor:								
No. required__________________	3	3	3	3	3	3	3	
Size____________________hp____	1/2	1/2	1/2	1/2	1/2	1/2	1/2	
Type of defrost__________________	Hot gas	Air	Air	Hot gas	Air	Air	Air	Hot
No. of units required_______________	2	2	2	2	1	6	2	
Condensing Unit:								
Model No________________________	CU-7½-12	CU-10-12	CU-15-12	CU-10-12	CU-7½-12	CU-20-22	CU-10-12	CU-1
No. of units required_____________	2	2	2	2	2	4	2	
Total operating kw______________	13 1	12 4	16 2	13 6	9.6	56 2	11.7	1

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		*° F.*	*In*		*° F.*	*In*	
B______	3.0	Expanded polystyrene.	50	5 0	Expanded polystyrene.	50	1.5	Expanded polystyrene
C______	3.5	______do______	45	5.5	______do______	45	2.5	Do.
D______	3 5	______do______	40	6 5	______do______	40	3 5	Do.
E______	5.0	Fibrous glass.	32	7 5	Fibrous glass	32	5.0	Do.
J______	5.0	______do______						
K______	5 0	Expanded polystyrene						
L______	5.5	______do______						
M______	8.0	Fibrous glass						
M-1____	4.5	____do______						

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3. Ceiling height is 20 feet in all areas except in the refrigerated spaces of firms 1 and 2, where height is 10 feet.
4. In the condensing unit model numbers, the first number is the horsepower of unit and the second number is the of refrigerant used in unit.

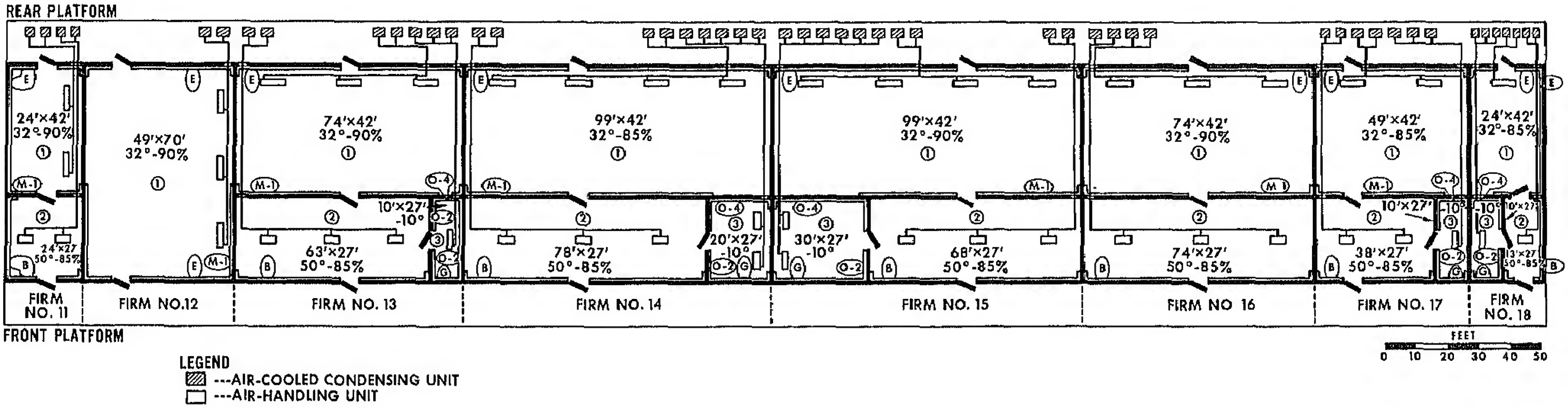


FIGURE 20.—Situation I, Building No. 2 (meat and meat products), floor plan and refrigeration equipment location.

REFRIGERATION AND EQUIPMENT SCHEDULE

Item	Firm 11		Firm 12	Firm 13			Firm 14			Firm 15			Firm 16		Firm 17			Firm 18		
	Room 1	Room 2	Room 1	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3	Room 1	Room 2	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3
Refrigeration																				
Carcass meat.....Pct..	100	---	75	75	---	---	50	---	---	85	---	---	100	---	---	---	---	---	---	---
Packaged meat...Pct	---	---	25	25	---	100	50	---	100	15	----	100	----	----	100	----	100	100	----	100
Refrigeration load B.t.u./hr..	33,685	33,799	78,147	74,170	68,870	77,074	98,024	78,288	139,024	96,324	73,248	170,347	74,170	75,851	69,070	40,100	76,606	42,415	27,801	76,600
Freezing capacity lb./hr.	---	---	---	---	---	430	---	---	800	---	---	1,200	---	---	---	---	430	---	---	430
Equipment																				
Air-handling unit:																				
Model No.	AH-8	AH-2	AH-9	AH-9	AH-3	AH-10	AH-9	AH-3	AH-10	AH-9	AH-3	AH-10	AH-9	AH-3	AH-9	AH-3	AH-10	AH-10	AH-3	AH-10
Rating B.t.u./hr./° F..	2,510	1,190	3,770	3,770	2,040	6,420	3,770	2,040	6,420	3,770	2,040	6,420	3,770	2,040	3,770	2,040	6,420	6,420	2,040	6,420
Air volume.....c.f.m..	4,950	1,515	7,480	7,480	2,270	12,500	7,480	2,270	12,500	7,480	2,270	12,500	7,480	2,270	7,480	2,270	12,500	12,500	2,270	12,500
Fan motor:																				
No. required	2	1	3	3	2	3	3	2	3	3	2	3	3	2	3	2	3	3	2	3
Size.........hp..	1/4	1/15	1/4	1/4	1/15	1/2	1/4	1/15	1/2	1/4	1/15	1/2	1/4	1/15	1/4	1/15	1/2	1/2	1/15	1/2
Type of defrost	Hot gas	Air	Hot gas	Hot gas	Air	Hot gas	Hot gas	Air	Hot gas	Hot gas	Air	Hot gas	Hot gas	Air	Hot gas	Air	Hot gas	Hot gas	Air	Hot gas
No. of units required	2	2	3	3	3	1	4	3	2	4	3	2	3	3	2	2	1	1	1	1
Condensing unit:																				
Model No.	CU-5-12	CU-3-12	CU-10-12	CU-7½-12	CU-7½-12	CU-10-22	CU-7½-12	CU-7½-12	CU-10-22	CU-7½-12	CU-7½-12	CU-10-22	CU-7½-12	CU-7½-12	CU-7½-12	CU-5-12	CU-10-22	CU-5-12	CU-2-12	CU-10-22
No. of units required	2	2	2	2	2	3	2	2	5	2	2	6	2	2	2	2	3	2	2	3
Total operating kw	5.7	4.9	12.3	11.6	9.4	22.8	12.9	10.6	43.8	12.7	10.0	40.3	11.6	10.3	11.0	5.8	22.0	6.0	3.9	22.0

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		*° F.*	*In.*		*° F.*	*In.*	
B	3.0	Expanded polystyrene.	50	5.0	Expanded polystyrene.	50	1.5	Expanded polystyrene
E	5.0	Fibrous glass	32	7.5	Fibrous glass.	32	5.0	Do.
G	10.5	do	−10	12.5	do	−10	10.0	Do.
M-1	4.5	do						
O-2	10.5	do						
O-4	8.5	do						

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3. 50° F. rooms are work areas for cutting, boning, packaging, and order assembly operations.
4. Ceiling height is 12 feet in all areas.
5. In the condensing unit model numbers, the first number is the horsepower of unit and the second number is the typ of refrigerant used in unit.

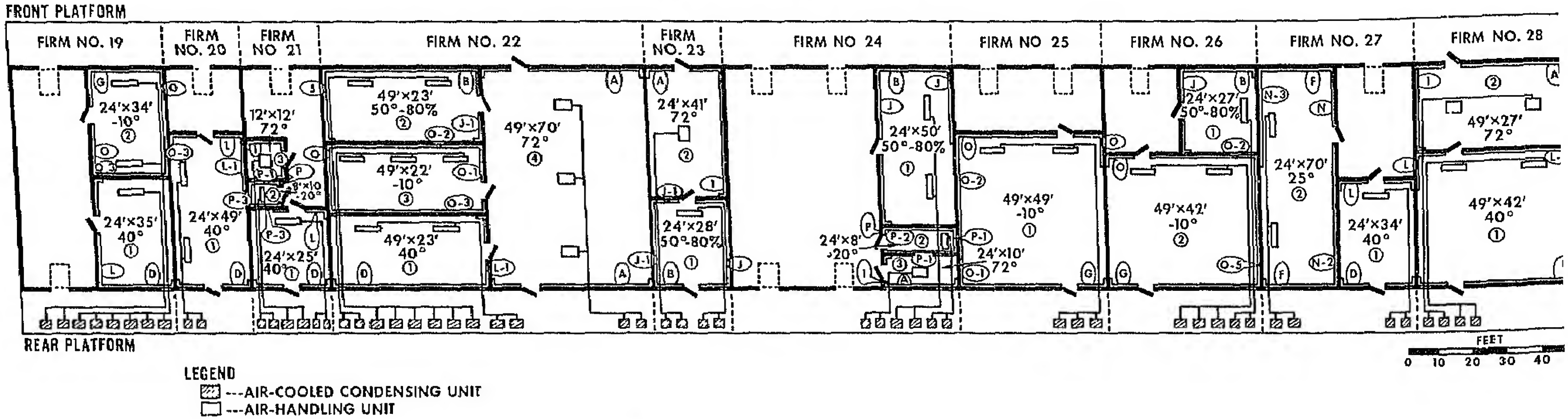


FIGURE 21.—Situation I, Building No. 3 (poultry and eggs), floor plan and refrigeration equipment location.

REFRIGERATION AND EQUIPMENT SCHEDULE

Item	Firm 19		Firm 20	Firm 21			Firm 22			
	Room 1	Room 2	Room 1	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3	Room 4
Refrigeration										
Refrigeration load B.t.u./hr.	38,768	159,787	53,483	30,831	15,012	9,901	53,922	64,158	196,934	81,045
Room usage or product handled.	Poultry	Poultry	Poultry	Poultry	Poultry	Cutting up poultry	Poultry	Shell eggs	Poultry	Order assembly
Freezing capacity lb/hr.		958			33				1,200	
Equipment										
Air-handling unit										
Model No.	AH-7	AH-10	AH-6	AH-6	AH-18	AH-1	AH-6	AH-5	AH-10	AH-3
Rating B.t.u./hr./°F	6,000	6,420	3,460	3,460	1,660	905	3,460	2,370	6,420	2,010
Air volume c.f.m.	10,500	12,500	6,500	6,500	3,000	1,170	6,500	4,300	12,500	2,270
Fan motor										
No. required	3	3	3	3	2	1	3	2	3	2
Size hp.	1/2	1/2	1/4	1/4	1/12	1/15	1/4	1/4	1/2	1/15
Type of defrost	Air	Hot gas	Air	Air	Hot gas	Air	Air	Air	Hot gas	Air
No. of units required	1	2	2	1	1	1	2	2	3	3
Condensing unit:										
Model No.	CU-3-12	CU-10-22	CU-7½-12	CU 3-12	CU-3-22	CU-½-12	CU-7½-12	CU-7½-12	CU-10-502	CU-5-12
No. of units required	2	6	2	2	2	2	2	2	6	2
Total operating kw.	5 9	50 7	7 8	5 0	4 7	1 6	7.7	9 0	45 4	9 7

Item	Firm 23		Firm 24			Firm 25	Firm 26		Firm 27		Firm 28	
	Room 1	Room 2	Room 1	Room 2	Room 3	Room 1	Room 1	Room 2	Room 1	Room 2	Room 1	Roo[m 2]
Refrigeration												
Refrigeration load B.t.u./hr.	37,695	32,693	57,589	33,160	8,620	56,685	39,953	69,531	40,364	72,921	70,103	42, 4
Room usage or product handled.	Shell eggs	Egg grading, order assembly	Shell eggs	Egg freezer	Egg breaking	Poultry	Shell eggs	Shell eggs	Poultry	Crusted chicken	Poultry	Orde[r] assem[bly]
Freezing capacity lb/hr.				141 4								
Equipment												
Air-handling unit												
Model No.	AH-6	AH-3	AH-5	AH 9	AH-1	AH-10	AH-6	AH-9	AH-7	AH-9	AH-7	AH-
Rating B.t.u./hr./°F	3,460	2,010	2,370	3,770	905	6,420	3,460	3,770	6,000	3,770	6,000	1,19
Air volume c.f.m.	6,500	2,270	4,300	7,480	1,170	12,500	6,500	7,480	10,500	7,480	10,500	1,51
Fan motor												
No. required	3	2	2	3	1	3	3	3	3	3	3	1
Size hp.	1/4	1/15	1/4	1/4	1/15	1/2	1/4	1/4	1/2	1/4	1/2	1/15
Type of defrost	Air	Air	Air	Hot gas	Air	Hot gas	Air	Hot gas	Air	Hot gas	Air	Air
No. of units required	1	1	2	1	1	2	1	2	1	2	2	2
Condensing unit:												
Model No.	CU-3-12	CU-2-12	CU-7½-12	CU-10-22	CU-½-12	CU-10-22	CU-5-12	CU-7½-22	CU-5-12	CU-7½-12	CU-7½-12	CU-3-1
No. of units required	2	2	2	2	2	3	2	3	2	2	2	2
Total operating kw.	5 7	3 5	8 0	11 2	1 4	26.8	6 1	21.0	6 1	10 8	11 2	6 8

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		°F.	In.		°F.	In.	
A	1.5	Expanded polystyrene.	72	4 5	Expanded polystyrene	72		None
B	3 0	do.	50	5.0	do.	50	1.5	Expanded polystyrene.
D	3.5	do.	40	5 5	do.	40	3 5	Do.
F	6.0	Fibrous glass	25	8 0	Fibrous glass.	25	5 5	Do
G	10.5	do.	−10	12 5	do.	−10	10.0	Do
I	2.0	Expanded polystyrene.	−20	13 0	do.	−20	11.0	Do
J	5 0	do.						
-1	4.0	do.						
[illegible]	5.5	do.						
-1	5 0	do.						
[illegible]	8 5	Fibrous glass.						
-2	4.0	do.						
-3	4 0	do.						

INSULATION SCHEDULE—Continued

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In		°F.	In		°F	In.	
O	13 0	do.						
O-1	12 5	do.						
O-2	10 5	do.						
O-3	9 5	do.						
O-5	8 5	do.						
P	13.5	do.						
P-1	13 0	do.						
P-2	11 0	do.						
P-3	10.5	do.						

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3 Ceiling height is 20 feet in all areas except in firms 21-2, 21-3, 24-2, and 24-3, where height is 10 feet.
4. In the condensing unit model numbers, the first number is the horsepower of unit and the second number is the type refrigerant used in unit.

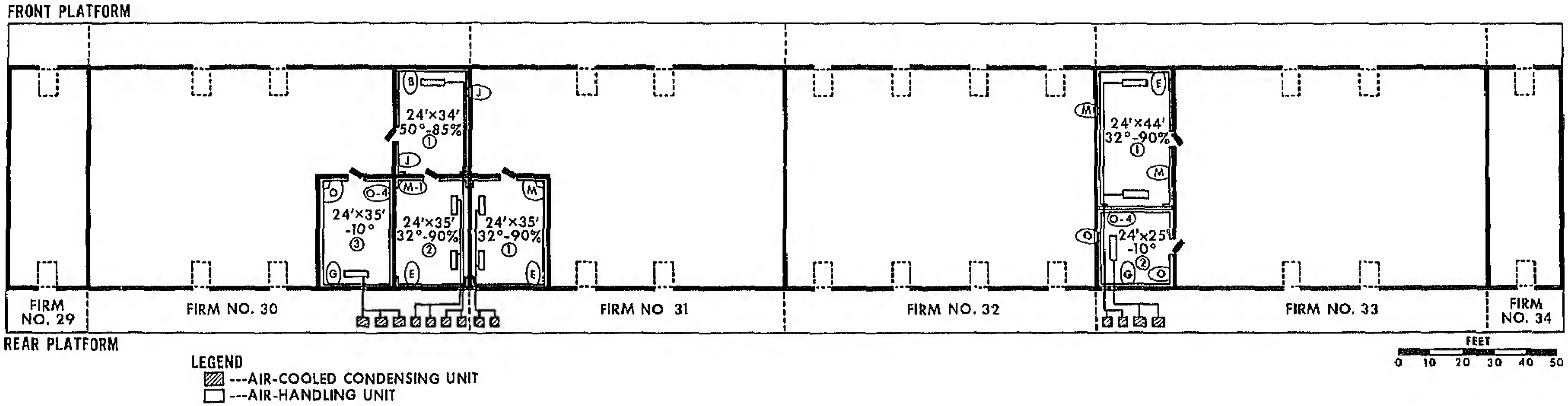


FIGURE 22.—Situation I, Building No. 4 (groceries with fruits and vegetables), floor plan and refrigeration equipment location.

REFRIGERATION SCHEDULE

Item	Firm 30 Room 1	Firm 30 Room 2	Firm 30 Room 3	Firm 31 Room 1	Firm 33 Room 1	Firm 33 Room 2
Refrigeration						
Refrigeration load B.t.u./hr.	73,976	75,883	47,508	77,886	86,024	34,434
Equipment						
Air-handling unit:						
Model No.	AH-7	AH-10	AH-9	AH-10	AH-10	AH-9
Rating...B.t.u./hr./° F.	6,000	6,420	3,770	6,420	6,420	3,770
Air volume...c.f.m.	10,500	12,500	7,480	12,500	12,500	7,480
Fan motor:						
No. required	3	3	3	3	3	3
Size...hp.	1/2	1/2	1/4	1/2	1/2	1/4
Type of defrost	Air	Hot gas	Hot gas	Hot gas	Hot gas	Hot gas
No. of units required	1	2	1	2	2	1
Condensing unit:						
Model No.	CU-7½-12	CU-7½-12	CU-5-22	CU-7½-12	CU-7½-12	CU-7½-22
No. of units required	2	2	3	2	2	2
Total operating kw.	10.3	11.5	13.4	11.8	13.2	9.8

INSULATION SCHEDULE

Code	Wall material Thickness	Wall material Type	Ceiling material Room temperature	Ceiling material Thickness	Ceiling material Type	Floor material Room temperature	Floor material Thickness	Floor material Type
	In.		*° F.*	*In.*		*° F.*	*In.*	
B	3.0	Expanded polystyrene.	50	6.0	Expanded polystyrene	50	1.5	Expanded polystyrene.
E	5.0	Fibrous glass.	32	7.5	Fibrous glass.	32	5.0	Do.
G	10.5	do	−10	12.5	do	−10	10.0	Do.
J	5.0	Expanded polystyrene.						
M	8.0	Fibrous glass.						
M-1	4.5	do						
O	13.0	do						
O-4	8.5	do						

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown
2. Humidities shown are minimum requirements.
3. Ceiling height is 20 feet in all areas
4. In the condensing unit model numbers, the first number is the horsepower of unit and the second number is the type refrigerant used in unit.

includes a plan, equipment schedule, refrigeration load schedule, and insulation schedule.

The insulation as shown on the drawing and as listed in the schedule was selected as described previously. The air-handling units and condensing units represent typical units that would meet each application.

An Alternate System Proposal

An alternate condensing unit selection is proposed on the basis of an absolute minimum cost. In most applications this means that only one condensing unit is selected per refrigerated space. When a firm has more than one refrigerated room, however, the same refrigerant can be used in both systems so that cross-connections can be made to provide partial refrigeration in case of emergency.

The alternate unit selections are not shown on the drawings, but are listed in the cost summaries and bills of materials to illustrate a cost comparison (tables 23 through 27).

The main disadvantage of the alternate proposal is the loss of refrigeration in case of compressor failure within the single condensing unit.

Equipment Costs

The interconnecting piping, piping insulation, and unit installation costs are proportioned between the air-handling units and condensing units. Each piece of equipment is complete as described under the introduction to Situation I and in the specifications.

Table 23 lists all firms, with their individual room-temperature requirements; the air unit and condensing unit selections, with the required quantity of each; and the total installed cost for the quantity of units required.

The alternate unit selections, with their installed costs, are also included in table 23. Note that the alternate selections affect only the condensing units, not the air-handling units. A still lower price is listed in the table for the alternate unit selection should semihermetic condensing units be chosen.

The following model number designations are used on the drawings and data sheets and in the table for both the original and alternate selections:

AH-7. AH = Air handling unit (evaporator unit) 7 = model number.
CU 5-12. CU = Air-cooled condensing unit 5 = Compressor horsepower. 12 = Type refrigerant.

Summary of all Costs

The total installed cost for the refrigeration equipment and insulation (including refrigeration doors) for Situation I is $931,598 for the base proposal and $882,972 for the alternate proposal. The total annual owning and operating cost for Situation I is $309,157 for the base proposal and $297,361 for the alternate p (table 12). The difference in cost between the base and alternate proposals minor in relation to the entire market. The low additional cost for the base p seems well justified, with its added protection against equipment failures and tial losses from product spoilage.

TABLE 12.—*Situation I, costs of refrigeration systems, by building*

[Air conditioning not included]

Expenses of refrigeration system	Building No. 1	2	3	4	c bu
Installed cost:					
Refrigeration equipment:	*Dollars*	*Dollars*	*Dollars*	*Dollars*	*I*
Base	142,667	184,269	161,209	47,531	5
Alternate	132,744	169,330	141,234	43,742	4
Insulation (including doors)	97,404	132,378	129,852	36,288	3
Base	240,071	316,647	291,061	83,819	9
Alternate	230,148	301,708	271,086	80,030	8
Annual owning and operating costs:					
Amortization of refrigeration equipment (10 yrs @ 6%):					
Base	19,386	25,037	21,903	6,458	7
Alternate	18,036	23,007	19,189	5,943	(
Maintenance of refrigeration equipment (10%/yr.)					
Base	14,267	18,427	16,121	4,753	5
Alternate	13,274	16,933	14,123	4,374	4
Amortization of insulation (20 yrs @ 6%)	8,492	11,541	11,320	3,164	3
Maintenance of insulation (2%/yr.)	1,948	2,648	2,598	726	
Insurance					
Base	435	573	527	152	
Alternate	417	546	491	145	
Taxes:					
Base	1,172	1,545	1,420	409	
Alternate	1,123	1,472	1,323	391	
Electric power cost	38,691	42,647	38,173	14,626	13
Base	84,389	102,418	92,062	30,288	30
Alternate	81,981	98,794	87,217	29,369	29

An actual worksheet used in calculating the insulation costs, power consumption, and electric power costs for each firm is included in the section on cost comparisons (table 22). Total costs, by firm, for refrigeration equipment, insulation, and owning and operating costs are listed in tables 24 through 27 of that section for Buildings 1, 2, 3, and 4, respectively.

SITUATION II, ONE CENTRAL SYSTEM FOR FOUR BUILDINGS

A central refrigeration system consists essentially of a single, relatively large, cooling equipment installation capable of supplying the necessary refrigerant to all the individual chilling and cold-storage areas. In Situation II, one central system handles all four buildings.

The various facilities described within the building complex have different refrigeration requirements, in regard not only to capacity, but also to temperature. Two different operating temperature ranges are required: −10° to −20° F. for freezers, and 32° and above for storage and chilling.

The basic design is a pump-feed liquid-ammonia recirculation system with an equipment building located at about an equal distance from all four buildings (fig. 23). Ammonia pumps are used to circulate the liquid to each of the four buildings at two different pressure levels, with each flow being metered both at the building and where it enters the individual firms. The amount of liquid fed into the evaporator is usually several times the amount that is actually evaporated in the coil and, therefore, liquid is always present in the suction return to the accumulators.

This type of system was chosen because (1) it simplifies the controls on the air-handling unit within the refrigerated space; (2) it allows better utilization of evaporator surface area, since refrigerant is in 100 percent of the coil; (3) superheat is not required; (4) the oil return is simpler than that of a direct-expansion system; and (5) a liquid-recirculation system is more economical than a brine system.

The office areas are heated in the winter and air conditioned in the summer, using the central refrigeration equipment. See the applicable part of the section on air conditioning and heating for a complete list of equipment selections and costs for Situation II and for deductions if air conditioning should be omitted.

Equipment Selection and Operation

The theory of operation and the components selected for the central system application can best be understood by reviewing the refrigerant-flow diagram (fig. 24). Figure 25 illustrates the physical location of this equipment within the central equipment building.

Compressor No. 5, rated at 30 horsepower, and compressor No. 6, rated at 75 horsepower, are the boosters that handle the low-stage space refrigeration loads. These two units, operating in parallel, provide 2 percent excess capacity.

Compressors Nos. 1, 2, and 3, rated at 200 horsepower each, and compressor No. 4, rated at 125 horsepower, are the high-stage units. They handle the high-stage space refrigeration load, the heat rejected by the low stage, and the air-conditioning load. These four units, operating in parallel, are 5.6 percent short of the total capacity; but since no diversity factor is used and the air-conditioning load is seasonal, they are acceptable.

Compressor No. 4 is piped up as a swing unit; that is, it can be used on the low stage in an emergency. If compressor No. 5 should become inoperative, 73 percent of the low-stage capacity could be maintained; or if compressor No. 6 should become inoperative, 100 percent of the low-stage capacity could be maintained. Duplicate controls for low-stage operation are required to make this compressor a swing unit.

If compressor No. 1, 2, or 3 should cease to operate, 68.4 percent of the high-stage load could be maintained; while, if compressor No. 4 should be shut down, 78.2 percent of the high-stage load could be maintained.

A separate compressor is not used for the air-conditioning system.

Standby pumps are included in the low- and the high-stage ammonia circuits and the air-conditioning/heating circuit. See table 33 for a description of all components.

The air-handling units are selected on a temperature differential of 7° to 10° F. except in the freezers, 72° rooms, and dry 50° rooms, where a maximum temperature differential of 15° is used. The selections are made for peak summer conditions and maximum product loading. Separate humidification equipment is not required when these temperature differentials are observed, since 85 to 90 percent relative humidity can be maintained in these short-term storage areas by the standard air-handling units selected at these specified temperature differentials.

Floor Plans and Air-Handling Equipment Layouts

Figures 26, 27, 28, and 29 illustrate the floor plans and air-handling equipment layouts for Buildings Nos. 1, 2, 3, and 4, respectively. The equipment schedule lists the air-handling units by model number, along with the operating characteristics of each type of unit. The source of refrigerant for the air-handling units must be traced back to the central equipment building, as illustrated in the plot plan (fig. 23) and the equipment building layout (fig. 25).

(Text continued on page 44)

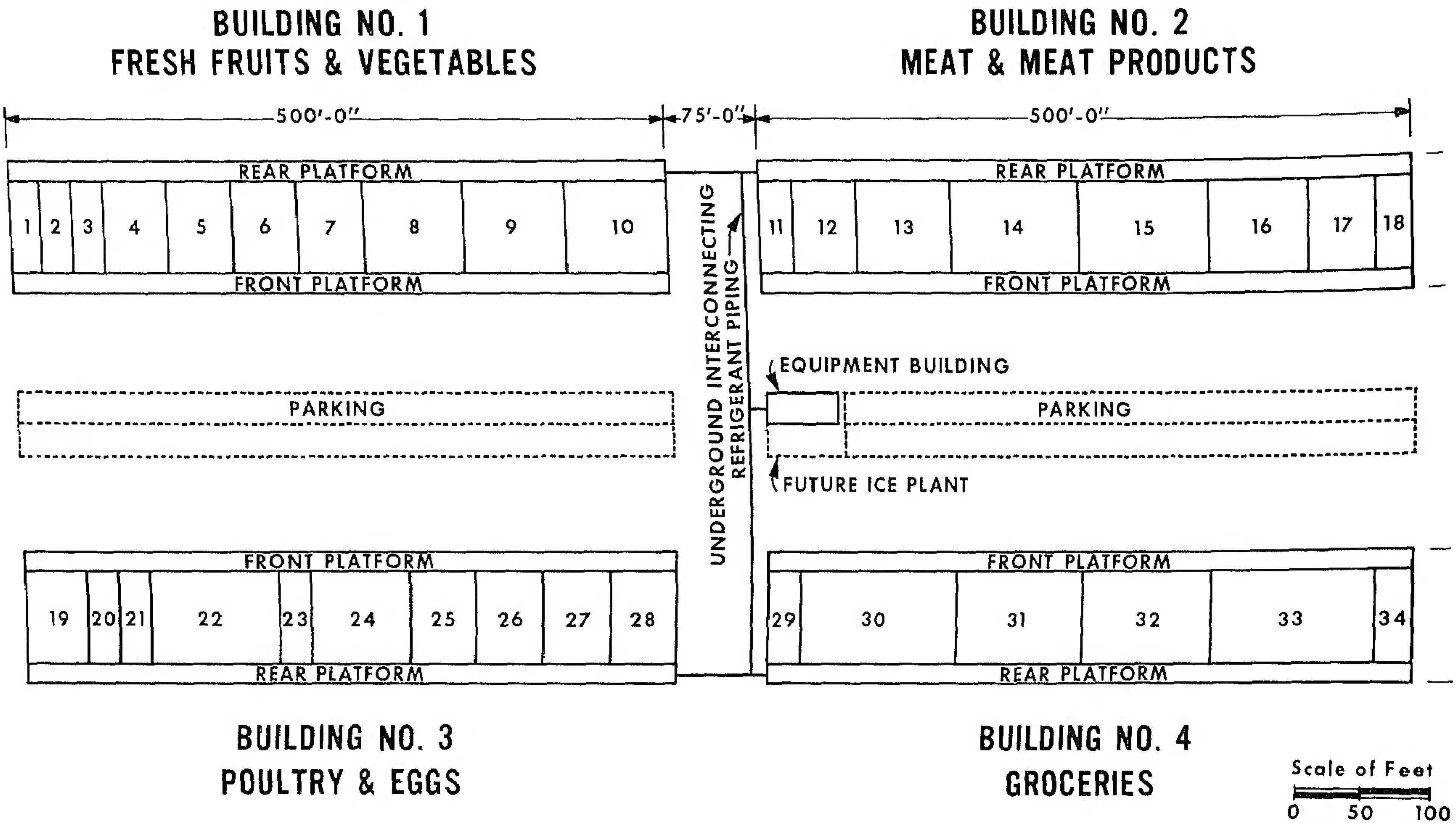


FIGURE 23.—Situation II, plot plan of food distribution center.

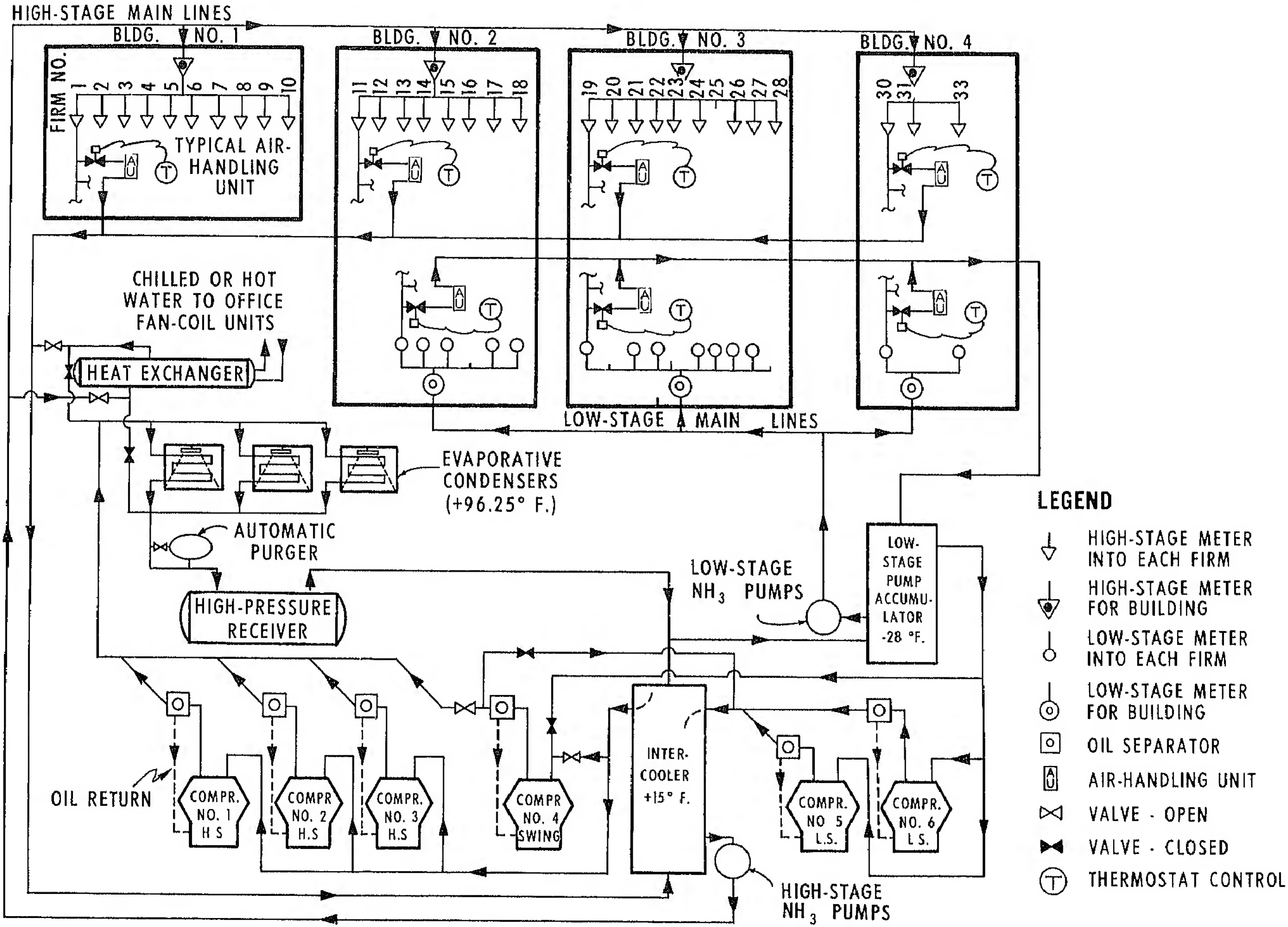


FIGURE 24.—Situation II, refrigerant flow diagram.

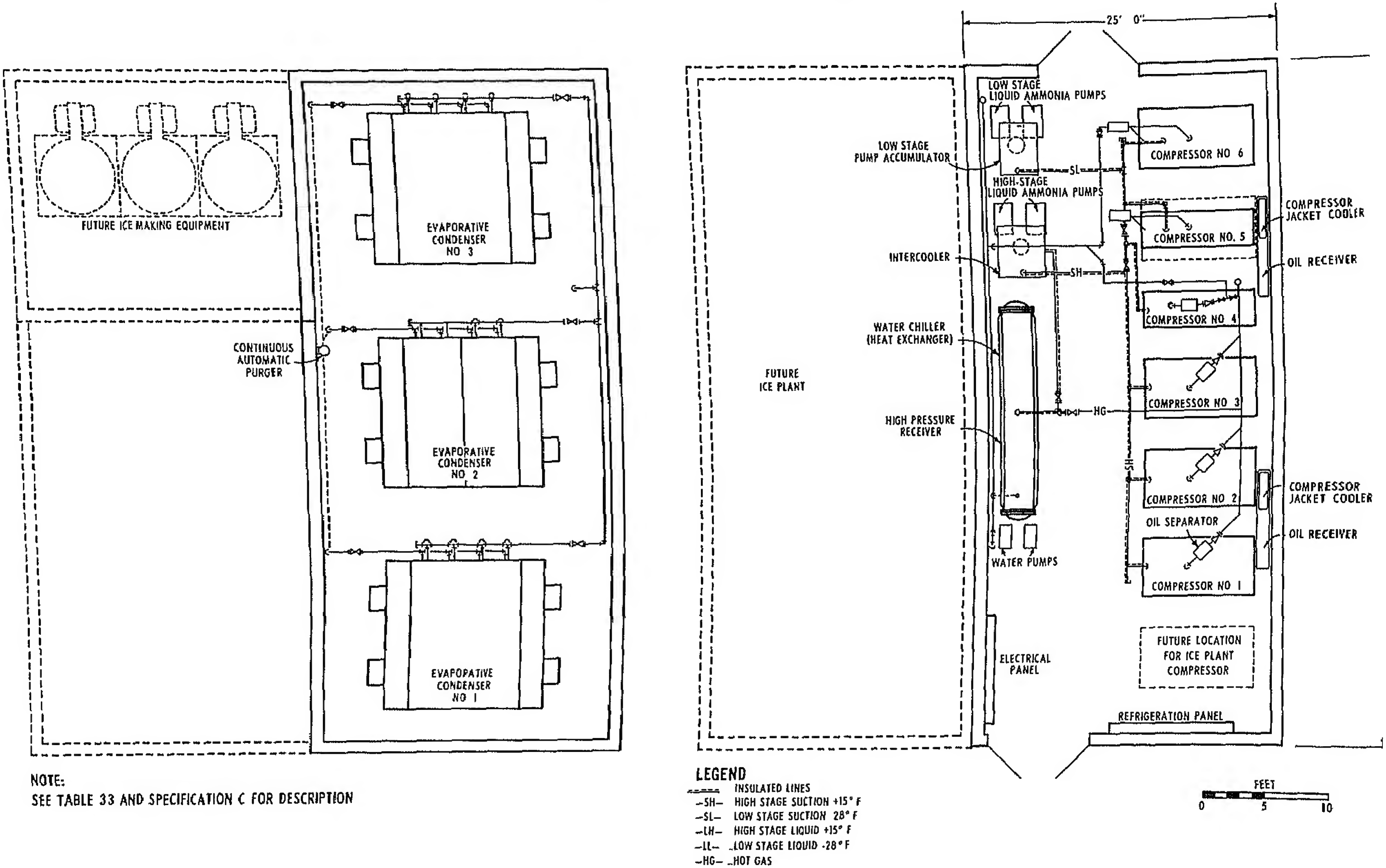


FIGURE 25.—Situation II, central system equipment building layout.

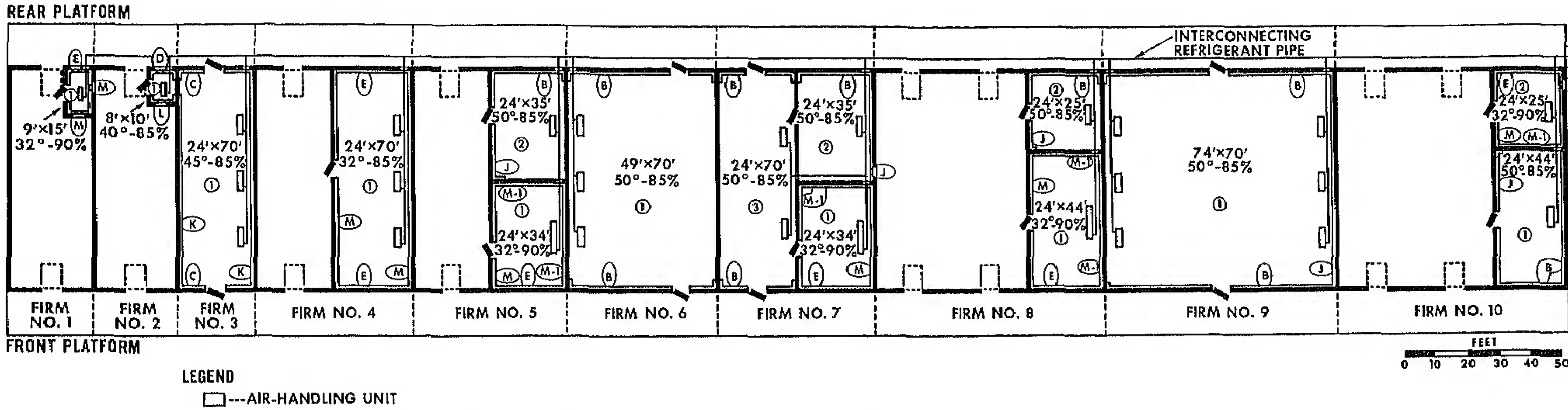


FIGURE 26.—Situation II, Building No. 1 (fresh fruits and vegetables), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 1	Firm 2	Firm 3	Firm 4	Firm 5		Firm 6	Firm 7			Firm 8		Firm 9	Firm 10	
	Room 1	Room 1	Room 1	Room 1	Room 1	Room 2	Room 1	Room 1	Room 2	Room 3	Room 1	Room 2	Room 1	Room 1	Room 2
Refrigeration load......B.t.u./hr..	17,800	10,532	160,705	175,370	95,034	103,780	302,555	95,034	103,780	140,550	111,705	72,864	480,780	111,692	71,570

EQUIPMENT SCHEDULE

Item	Model					
	AH-2RX	AH-5RX	AH-6RX	AH-7RX	AH-13RX	AH-14RX
Air handling unit:						
Rating B.t.u./° F..	1,430	2,960	4,330	7,500	7,500	12,500
Air volume c.f.m..	1,515	4,300	6,500	10,500	11,000	18,300
Fan:						
No. required...	1	2	3	3	3	5
Size.........in..	16	16	16	18	18	18
Fan motor:						
No. required...	1	2	3	3	3	5
Size.......hp..	1/15	1/4	1/4	1/2	1/2	1/2
Type of defrost...	Air	Air	Air	Air	Hot gas	Hot gas
No. of units required	1	1	2	16	4	3

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		*° F.*	*In.*		*° F.*	*In.*	
B......	2.0	Expanded polystyrene.	50	3.0	Expanded polystyrene.	50	1.0	Expanded polystyrene.
C......	2.0	......do......	45	3.5	......do......	45	1.5	Do.
D......	2.0	......do......	40	3.5	......do......	40	2.0	Do.
E......	2.5	......do......	32	3.5	......do......	32	2.5	Do.
J......	3.0	......do......						
K......	3.0	......do......						
L......	3.5	......do......						
M......	3.5	......do......						
M-1....	2.0	......do......						

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3. Ceiling height is 20 feet in all areas except in the refrigerated spaces of firms 1 and 2 where height is 10 feet.
4. The "RX" designation in the air-handling unit model means "recirculated liquid ammonia."

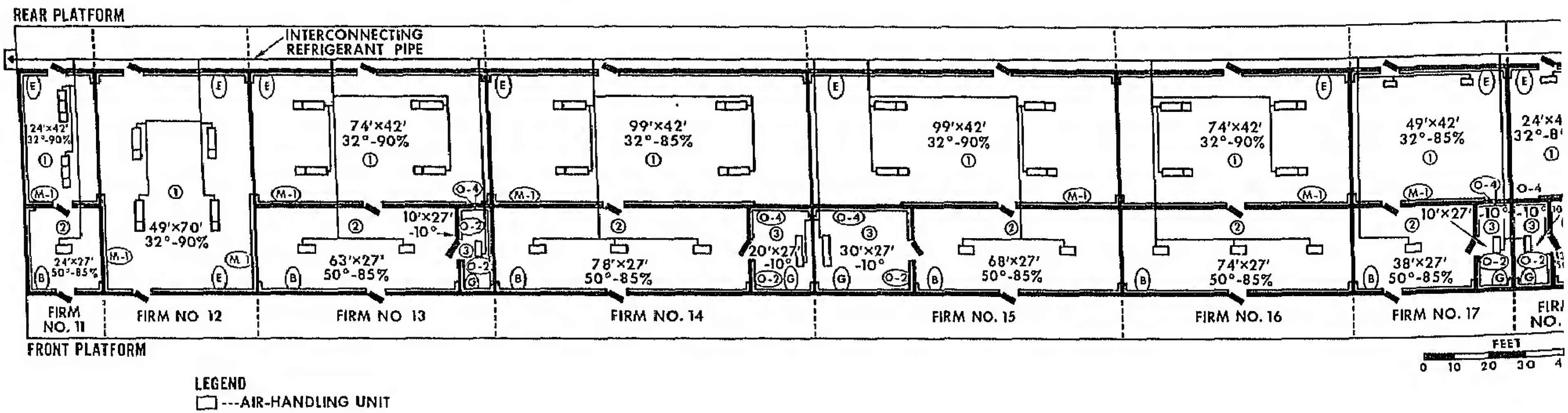


FIGURE 27.—Situation II, Building No. 2 (meat and meat products), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 11		Firm 12	Firm 13			Firm 14			Firm 15			Firm 16		Firm 17			Firm 18		
	Room 1	Room 2	Room 1	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3	Room 1	Room 2	Room 1	Room 2	Room 3	Room 1	Room 2	I
Carcass meat......Pct..	100	----------	75	75	----------	----------	50	----------	----------	85	----------	----------	100	----------	----------	----------	----------	----------	----------	
Packaged meat...Pct..	----------	----------	25	25	----------	100	50	----------	100	15	----------	100	----------	----------	100	----------	100	100	----------	
Refrigeration load B.t.u /hr..	37,062	36,337	98,427	86,945	74,492	74,731	123,065	85,210	139,821	120,365	79,280	175,079	86,945	82,003	79,303	43,378	78,114	46,942	20,364	
Freezing capacity lb./hr..	----------	----------	----------	----------	----------	430	----------	----------	860	----------	----------	1,290	----------	----------	----------	----------	430	----------	----------	

EQUIPMENT SCHEDULE

Item	Model							
	AH-3RX	AH-11RX	AH-12RX	AH-13RX	AH-14RX	AH-15RX	AH-16RX	AH-17RX
Air-handling unit								
Rating								
B.t.u./hr./° F..	2,040	3,140	4,730	7,500	12,500	2,150	2,500	2,850
Air volume....c.f.m..	2,270	4,050	7,450	11,000	18,300	2,300	2,900	3,700
Fan.								
No. required.......	2	2	3	3	5	[1] 1	[1] 1	[1] 1
Size.............in..	14	16	16	18	18	----------	----------	----------
Fan motor								
No. required.......	2	2	3	3	5	1	1	1
Size............hp..	1/15	1/4	1/4	1/2	1/2	1/4	1/2	1
Type of defrost.......	Air	Hot gas	Hot gas	Hot gas	Hot gas	Hot gas	Hot gas	Hot gas
No. of units required...	13	2	1	3	2	10	4	8

[1] Centrifugal

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	T
	In		° F	In.		° F.	In.	
B......	2.0	Expanded polystyrene.	50	3.0	Expanded polystyrene.	50	1.0	Expan polys
E......	2.5	......do......	32	3.5	......do......	32	2.5	D
G......	5.5	Fibrous glass.	−10	6.5	Fibrous glass.	−10	5.5	D
M-1....	2.0	Expanded polystyrene	----------	----------	----------	----------	----------	----------
O-2.....	5.5	Fibrous glass.	----------	----------	----------	----------	----------	----------
O-4.....	4.5	......do......	----------	----------	----------	----------	----------	----------

NOTES.
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements
3. 50° F. rooms are work areas for cutting, boning, packaging, and order assembly operations.
4. Ceiling height is 12 feet in all areas.
5 The "RX" designation in the air-handling model numbers means "recirculated liquid ammonia."

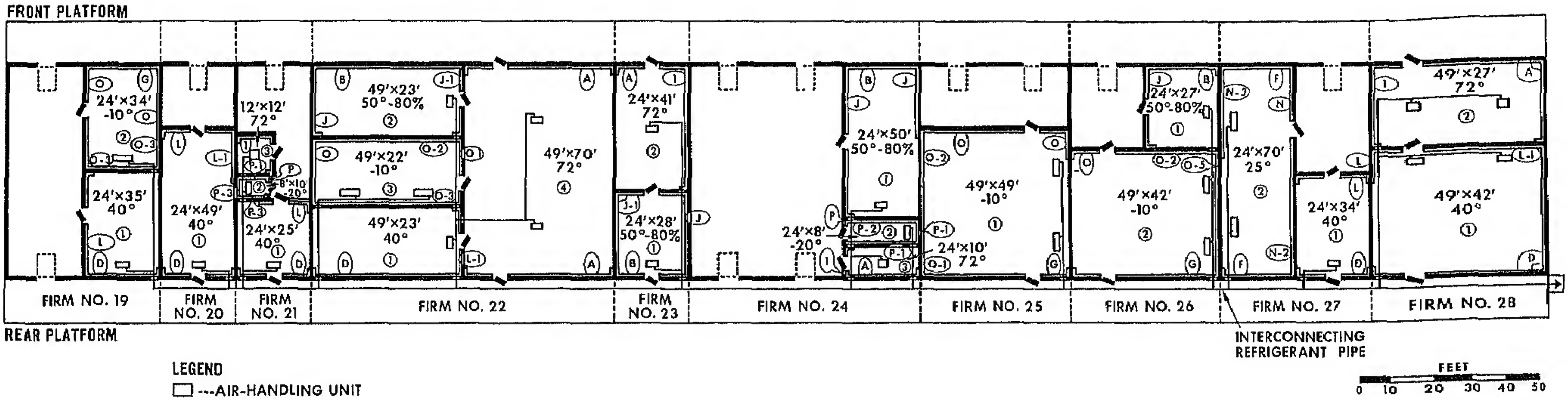


FIGURE 28.—Situation II, Building No. 3 (poultry and eggs), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 19		Firm 20	Firm 21			Firm 22				Firm 23		Firm 24			Firm 25	Firm 26		Firm 27		Firm 28	
	Room 1	Room 2	Room 1	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3	Room 4	Room 1	Room 2	Room 1	Room 2	Room 3	Room 1	Room 1	Room 2	Room 1	Room 2	Room 1	Room 2
Refrigeration load B.t.u./hr..	42,814	105,541	58,538	33,710	15,750	10,338	50,404	69,093	203,355	92,705	71,102	35,005	62,148	34,402	9,705	98,440	44,522	79,340	44,328	84,607	83,145	46,131
Room usage or product handled	Poultry	Poultry	Poultry	Poultry	Poultry	Cutting up poultry	Poultry	Shell eggs	Poultry	Order assembly	Shell eggs	Egg grading, order assembly	Shell eggs	Egg freezer	Egg breaking	Poultry	Shell eggs	Poultry	Poultry	Crusted chicken	Poultry	Order assembly
Freezing capacity lb./hr.		958			38				1,200					141 4						1,228		

EQUIPMENT SCHEDULE

Item	Model							
	AH-2RX	AH-3RX	AH-6RX	AH-7RX	AH-11RX	AH-12RX	AH-13RX	AH-14RX
Air-handling unit:								
Rating B.t.u./hr./° F..	1,430	2,040	4,330	7,500	3,140	4,730	7,500	12,500
Air volume.....c.f.m..	1,515	2,270	6,500	10,500	4,950	7,450	11,000	18,300
Fan:								
No. required	1	2	3	3	2	3	3	5
Size..........in..	16	14	16	18	16	16	18	18
Fan motor:								
No. required	1	2	3	3	2	3	3	5
Size..........hp..	1/15	1/15	1/4	1/2	1/4	1/4	1/2	1/2
Type of defrost	Air	Air	Air	Air	Hot gas	Hot gas	Hot gas	Hot gas
No. of units required	4	3	4	7	3	2	5	1

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3. Ceiling height is 20 feet in all areas except in firms 21–2, 21–3, 24–2, and 24–3, where height is 10 feet.
4. The "RX" designation in the air-handling unit model means "recirculated liquid ammonia"

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		° F.	In.		° F.	In.	
A	0.5	Expanded polystyrene.	72	3.0	Expanded polystyrene.	72		None
B	2.0	do.	50	3.0	do.	50	1.0	Expanded polystyrene
D	2.0	do.	40	3.5	do.	40	2.0	Do.
F	3.0	do.	25	4.0	do.	25	3.0	Do.
G	5.5	Fibrous glass.	−10	6.5	Fibrous glass.	−10	5.5	Do.
I	1.0	Expanded polystyrene.	−20	7.0	do.	−20	6.0	Do.
J	3.0	do.						
J-1	2.5	do.						
L	3.5	do.						
L-1	3.0	do.						
N	4.0	do.						
N-2	2.0	do.						
N-3	3.0	do.						
O	7.0	Fibrous glass						
O-1	6.5	do.						
O-2	5.5	do.						
O-3	5.0	do.						
O-5	4.5	do.						
P	7.5	do.						
P-1	7.0	do.						
P-2	6.0	do.						
P 3	5.5	do.						

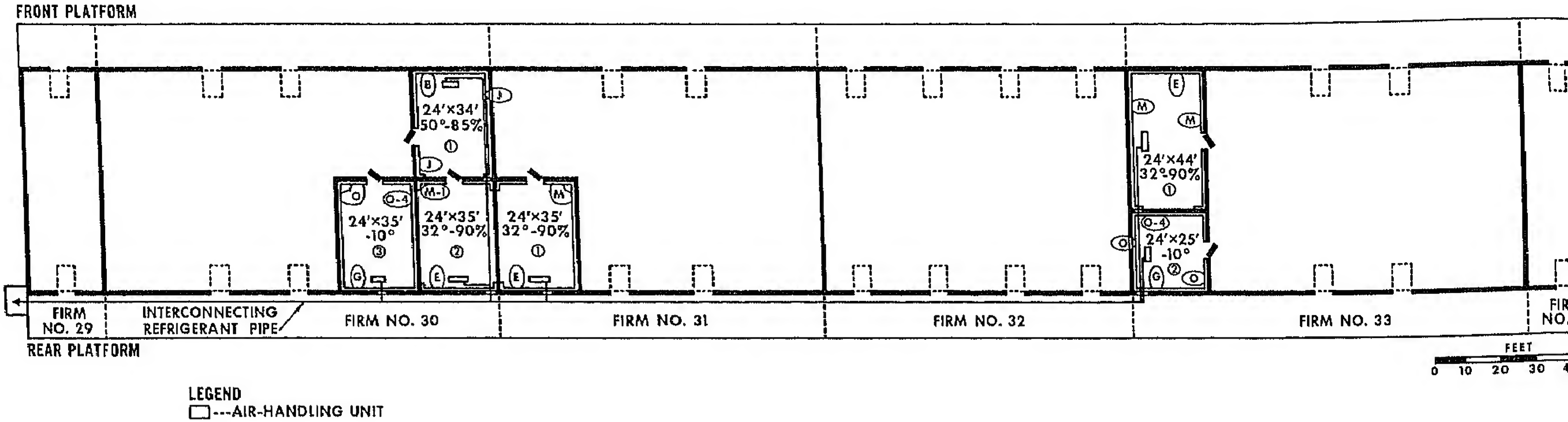


FIGURE 29.—Situation II, Building No. 4 (groceries with fruits and vegetables), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 30			Firm 31	Firm 33	
	Room 1	Room 2	Room 3	Room 1	Room 1	Room 2
Refrigeration load B.t.u /hr..	77,865	81,611	52,260	85,833	95,368	38,537

EQUIPMENT SCHEDULE

Item	Model		
	AH-7RX	AH-12RX	AH-14RX
Air-handling unit:			
Rating ---------- B.t.u./° F..	7,500	4,730	12,500
Air volume ---------- c.f.m..	10,500	7,450	18,300
Fan:			
No. required ----------	3	3	5
Size ---------- in..	18	16	18
Fan motor:			
No. required ----------	3	3	5
Size ---------- hp..	1/2	1/4	1/2
Type of defrost ----------	Air	Hot gas	Hot gas
No. of units required ----------	1	2	3

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown
2. Humidities shown are minimum requirements.
3. Ceiling height is 20 feet in all areas.
4. The "RX" designation in the air-handling model numbers means "recirculated liquid ammonia."

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Ty
	In.		° F	In		° F	In.	
B ------	2 0	Expanded polystyrene	50	3 0	Expanded polystyrene.	50	1 0	Expande polysty
E ------	2.5	------do------	32	3 5	------do------	32	2.5	Do.
G ------	5 5	Fibrous glass.	-10	6 5	Fibrous glass.	-10	5.5	Do.
J ------	3.0	Expanded polystyrene						
M ------	3 5	------do------						
M-1 ----	2 0	------do------						
O ------	7 0	Fibrous glass						
O-4 ----	4.5	------do------						

Equipment Costs

The cost of the interconnecting piping, with its insulation and installation costs, is proportioned between the air-handling units and the central engine room.

Table 33 lists all the equipment used in the central-system equipment building, along with unit costs. Table 34 lists the air-handling equipment and unit costs for the four buildings.

Summary of all Costs

The total installed cost for the refrigeration equipment and insulation (includ refrigeration doors) for Situation II is $893,277. The total annual owning a operating cost for Situation II is $190,941 (table 13).

Information used to arrive at these final cost figures is included in tables through 31 showing insulation costs and power consumption (kw.-hr./yr.) caus by transmission heat gains in Buildings Nos. 1, 2, 3, and 4. Table 32 show tabulation of power consumption and electric power costs in Buildings Nos. 1, 2, and 4.

TABLE 13.—*Situation II, installed costs and annual owning and operating costs for all buildings*

Expenses of refrigeration system[1]	Labor cost	Installed cost including labor
Equipment and insulation costs:	*Dollars*	*Dollars*
Air-handling units	48,254	[2]197,645
Pipe insulation (outside engine room)	17,010	51,020
Central engine room and piping	24,700	[2]170,800
Pipe and shell insulation (in engine room)	6,667	20,000
Air conditioning (four office areas)	21,480	60,888
Pipe insulation (air conditioning)	3,470	10,400
Cold-storage room insulation	111,994	265,389
Cold-storage room doors		78,885
Refrigerant metering devices		38,250
Total installed cost	233,575	[3]893,277

Annual owning and operating costs:		Total cost
Amortization, capital cost (20 yr. @ 6%)	893,277 × 0 08718 =	77,876
Maintenance, insulation (2%/yr.)	344,274 × 0.02 =	6,885
Maintenance, refrigeration, on contract basis		20,500
Maintenance, air conditioning, on contract basis		2,688
Insurance, $1.81/thousand adjusted	893,277 × 0.00181 =	1,617
Taxes, $4.88/thousand adjusted	893,277 × 0.00488 =	4,359
Electric power cost		77,016
Total annual owning and operating cost		190,941

[1] Refrigeration load, high-stage 358.6 TR.
Refrigeration load, low-stage 103.3 TR.
Air-conditioning load 120.0 TR.

[2] Includes proportionate share of interconnecting piping costs

[3] If office areas are not to be air conditioned, $18,620 can be deducted from this figure, and related owning and operating costs will be lower.

SITUATION III, ONE CENTRAL SYSTEM FOR EACH OF FOUR BUILDINGS

In Situation II, one central refrigeration system was proposed to handle all four buildings; but in Situation III, one central refrigeration system is proposed for each of the four main buildings.

The basic system design for Buildings Nos. 1, 2, and 3 is again a pump-feed liquid-ammonia recirculation system with ammonia pumps that circulate the liquid throughout each of the buildings at two different pressure levels. The refrigerant flow is metered where it leaves the central equipment room and again where it enters each individual firm.

The refrigeration system for Building No. 4 is designed as a direct-expansion ammonia system because of the small refrigeration requirement. The total load in this building is 27.7 tons high stage, 7.2 tons low stage, and 30 tons of air conditioning.

All central equipment rooms are located in spaces 9 feet high underneath the rear platforms (fig. 30).

The office areas are heated in the winter and air conditioned in the summer by the central refrigeration equipment. See the applicable part of the section on air conditioning and heating for equipment selection and costs for Situation III, and for deductions if air conditioning should be omitted.

Equipment Selection, Operation, and Cost Summary

The air-handling units are selected for a temperature differential of 7° to 9° F., except in the freezers, 72° rooms, and dry 50° rooms, where a maximum of 15° is used. Selections are based on peak summer conditions and maximum product loading. Separate humidification equipment is not required, because 85 to 90 percent relative humidity can be maintained in these short-term storage areas with the standard air-handling units as selected. The refrigeration loads within the conditioned spaces are slightly less than in Situation II because of the increased thickness of insulation in certain areas.

The evaporative condensers are installed inside the equipment room under the rear platform. They exhaust air that is drawn through the equipment room, thus helping to ventilate this room in the summer. During winter, a good part of the condenser heat is used to heat the office areas.

Separate compressors and condensers are not used for air conditioning and heating duty, because it is more economical to incorporate these requirements into the refrigeration-equipment selections.

The cost of the interconnecting piping, with its insulation and installation costs, is proportioned between the air-handling units and the central engine room.

Tables 43 through 46, in the section on cost comparisons, list all the equipment used in each building, along with unit costs.

Building No. 1 (Fresh fruits and vegetables)

To understand the theory of operation and the components selected for Building No. 1, see the refrigerant-flow diagram (figure 31), and the physical layout of the central-system components (figure 32).

Building No. 1 has only a high-stage load and air-conditioning load, since all

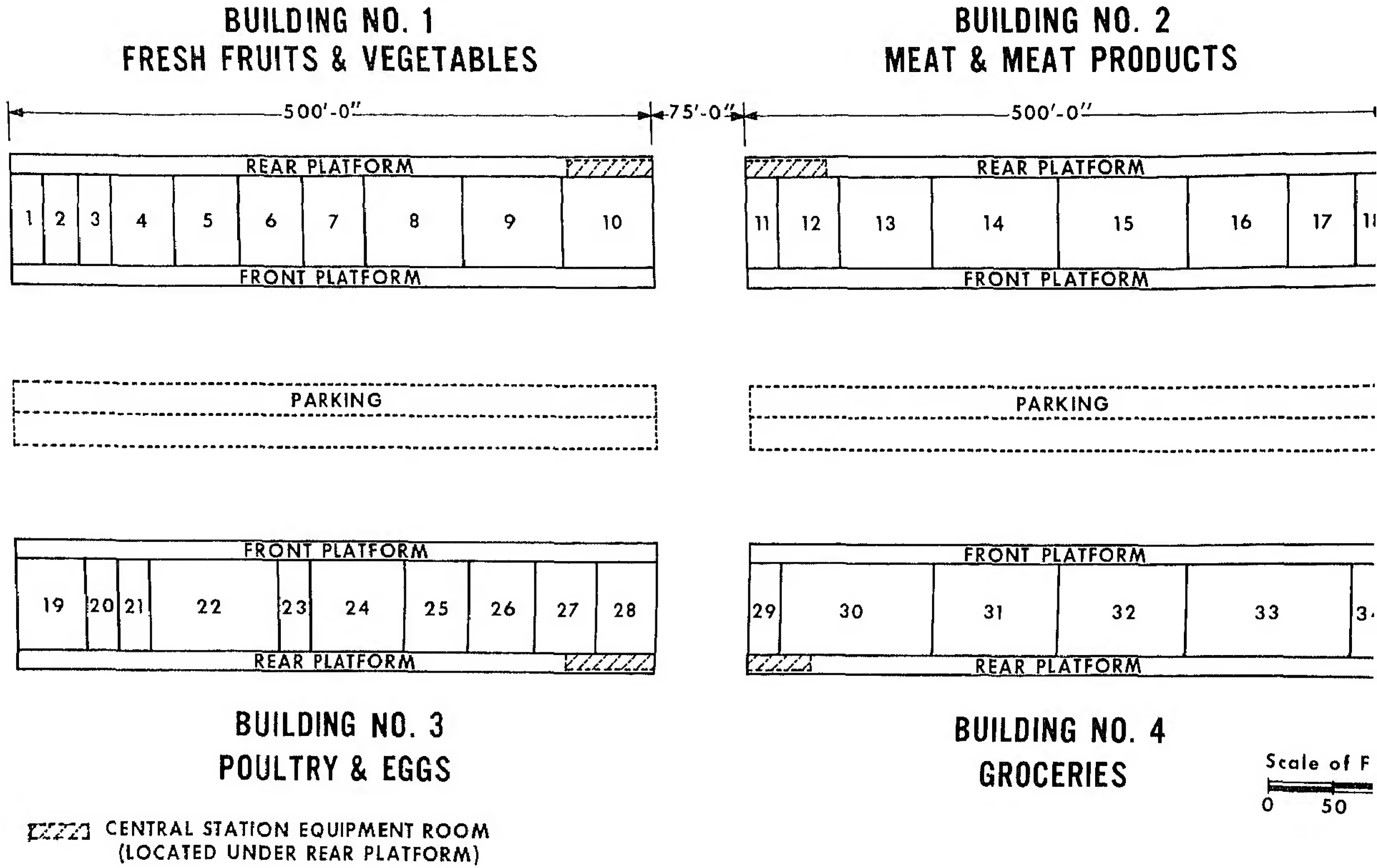


FIGURE 30.—Situation III, plot plan of food distribution center.

storage spaces are 32° F. and above. The four compressors operating in parallel fall 5 percent short of meeting the maximum load requirements. However, a diversity factor was not used and the air-conditioning load is seasonal, which makes the balance between these four units acceptable.

If compressor No. 1 or 3 should fail to operate, 75 percent of the load c maintained by the other three compressors; or, if compressor No. 2 or 4 become inoperative, 69 percent of the capacity could still be maintained.

A standby liquid-ammonia pump and water pump are provided.

The floor plan for building No. 1 is shown in figure 33.

Table 14 gives a summary of all installed costs and owning and operating costs necessary to meet the refrigeration requirements for the fruit and vegetable dealers.

Other appropriate information used to arrive at these final cost figures is included in the section on cost comparisons. Table 35 shows for Situation III, Building No. 1, insulation costs and power consumption (kw. hrs./yr.) caused by transmission-heat gains. Table 39 shows power consumption and electric power costs. Table 43 is the bill of materials and unit cost.

Building No. 2 (Meats and meat products)

The refrigerant-flow diagram (fig. 34) and the central equipment room illustration (fig. 35) show the operation and components selected to meet the refrigeration requirements of Building No. 2.

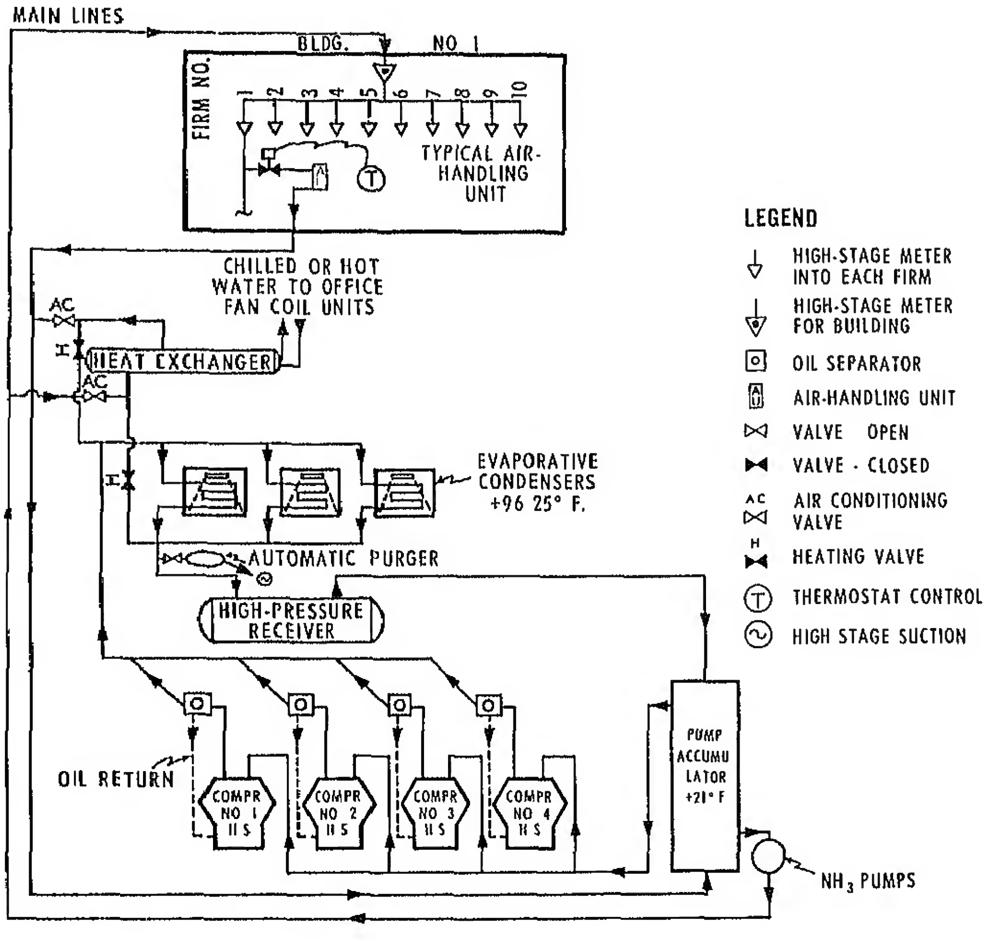


Figure 31.—Situation III, Building No. 1, refrigerant-flow diagram.

TABLE 14.—*Situation III, Building No. 1, installed costs and annual owning and operating costs*

Expenses of refrigeration system[1]	Labor cost	Installed cost including labor
Equipment and insulation costs	*Dollars*	*Dollars*
Air-handling units	11,050	[2]31,665
Pipe insulation (outside of engine room)	2,935	8,900
Central engine room and piping	6,370	[2]53,430
Pipe and shell insulation (in engine room)	1,335	4,000
Air conditioning (office area)	5,370	15,222
Pipe insulation (air conditioning)	865	2,600
Cold-storage room insulation	33,057	74,019
Cold-storage room doors		17,600
Refrigerant metering devices		8,090
Total installed cost	60,982	[3]235,526
Annual owning and operating costs		*Total cost*
Amortization, capital cost (20 yr. @ 6%)	235,526 × 0.08718 =	20,533
Maintenance, insulation (2%/yr.)	91,619 × 0.02	1,832
Maintenance, refrigeration, on contract basis		8,565
Maintenance, air conditioning, on contract basis		672
Insurance, $1.81/thousand adjusted	235,526 × 0.00181 =	426
Taxes, $4.88/thousand adjusted	235,526 × 0.00488 =	1,149
Electric power cost		29,364
Total annual owning and operating cost		62,541

[1] Refrigeration load 167.9 TR.
Air conditioning load 30.0 TR

[2] Includes proportionate share of interconnecting piping costs.

[3] If office areas are not to be air conditioned, $5,180 can be deducted from this figure, and related owning and operating costs will be lower.

A pump-feed liquid-ammonia recirculation system is used to handle the low- and high-stage refrigeration pads. Two ammonia pumps and one water pump are provided for standbys.

Compressors Nos. 4 and 5, booster compressors operating in parallel, are 6 percent short of meeting the full-load low-stage capacity; but a diversity factor was not used, which makes this combination acceptable.

Compressors Nos. 1 and 2 handle the high-stage space refrigeration load, the air-conditioning load, and the heat rejected by the booster compressors. These two compressors provide a capacity slightly in excess of the maximum that would be required at full load.

Compressor No. 3 is a swing unit capable of operating on either the low or high stages. If one of the high-stage compressors should become inoperative, a minimum of 74 percent of the high-stage capacity could still be maintained. If one of the

(Text continued on page 52)

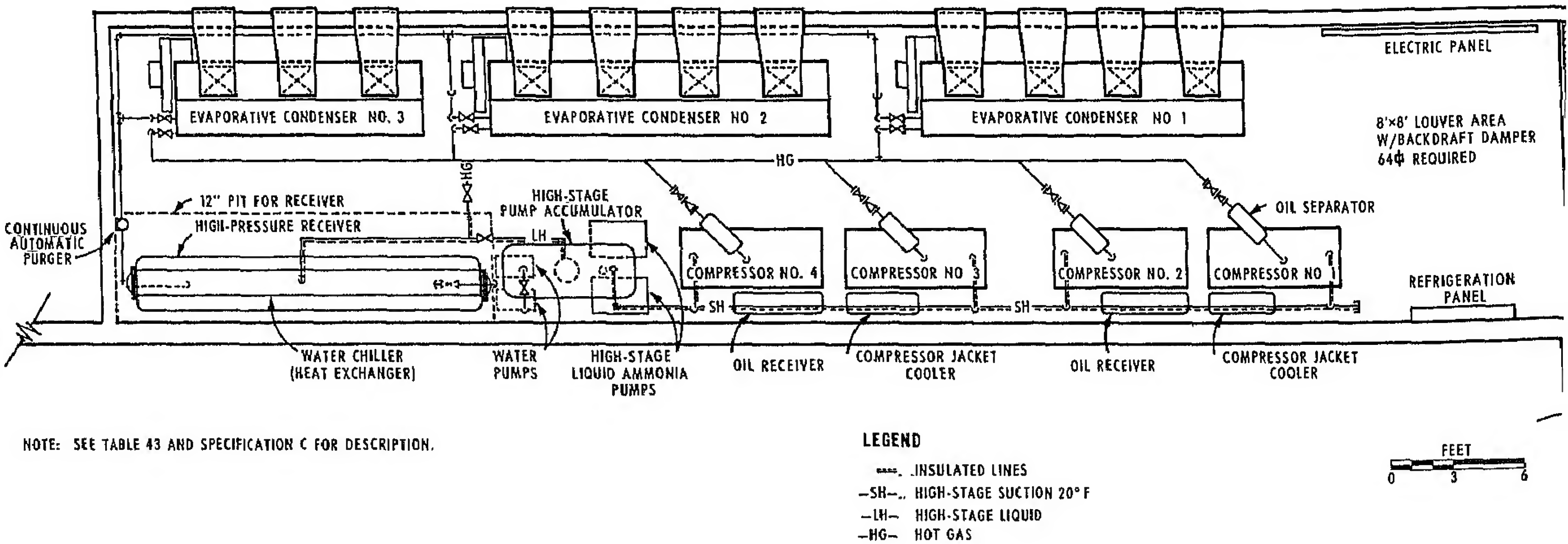


Figure 32.—Situation III, Building No. 1, central-system equipment room.

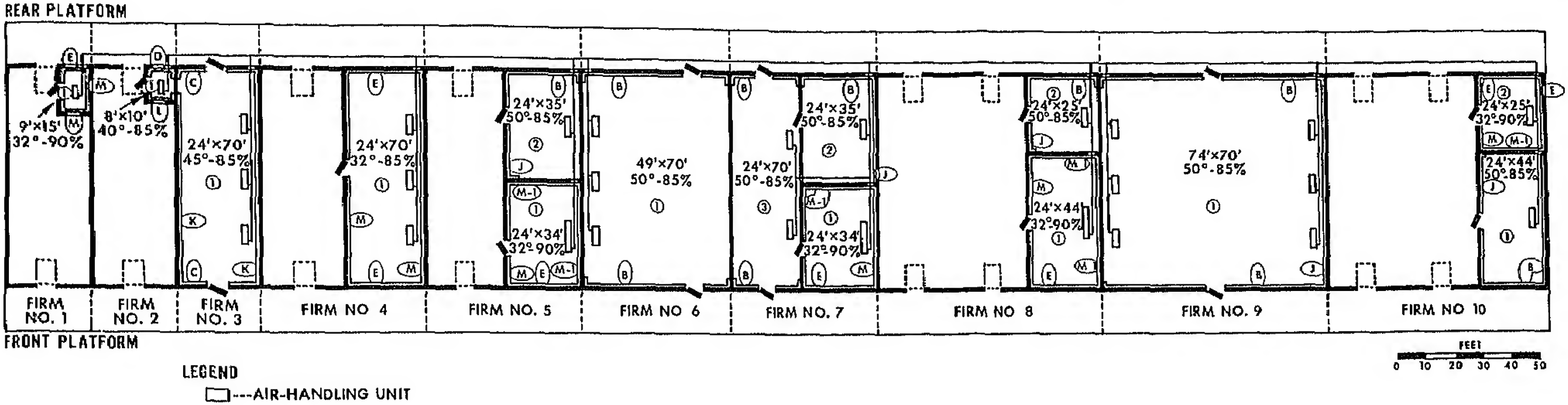


FIGURE 33.—Situation III, Building No. 1 (fresh fruits and vegetables), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 1	Firm 2	Firm 3	Firm 4	Firm 5		Firm 6	Firm 7			Firm 8		Firm 9	Firm 10	
	Room 1	Room 1	Room 1	Room 1	Room 1	Room 2	Room 1	Room 1	Room 2	Room 3	Room 1	Room 2	Room 1	Room 1	Room 2
Refrigeration load...B.t.u./hr..	17,701	10,337	157,495	174,230	93,107	102,320	297,255	93,107	102,320	137,810	109,215	71,819	472,520	108,833	69,791

EQUIPMENT SCHEDULE

Item	Model					
	AH-2RX	AH-5RX	AH-6RX	AH-7RX	AH-13RX	AH-14RX
Air-handling unit:						
Rating...B.t.u./hr./° F..	1,430	2,000	4,330	7,500	7,500	12,500
Air volume........c.f.m..	1,515	4,300	6,500	10,500	11,000	18,300
Fan:						
No. required...........	1	2	3	3	3	5
Size.................in..	16	16	16	18	18	18
Fan motor.						
No. required...........	1	2	3	3	3	5
Size.................hp..	1/15	1/4	1/4	1/2	1/2	1/2
Type of defrost...........	Air	Air	Air	Air	Hot gas	Hot gas
No. of units required.......	1	1	2	10	4	3

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In		*° F.*	*In.*		*° F.*	*In.*	
B......	2.0	Expanded polystyrene.	50	4.0	Expanded polystyrene.	50	1.0	Expanded polystyrene.
C......	2.5	......do......	45	4.0	......do......	45	2.0	Do.
D......	2.5	......do......	40	4.0	......do......	40	2.5	Do.
E......	3.0	......do......	32	4.0	......do......	32	3.0	Do.
J......	3.5	......do....						
K......	3.5	......do....						
L......	4.0	......do....						
M......	4.5	......do....						
M-1....	2.5	......do....						

NOTES.
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3. Ceiling height is 20 feet in all areas except in the refrigerated spaces of firms 1 and 2 where height is 10 feet.
4. The "RX" designation in the air-handling unit model means "recirculated liquid ammonia."

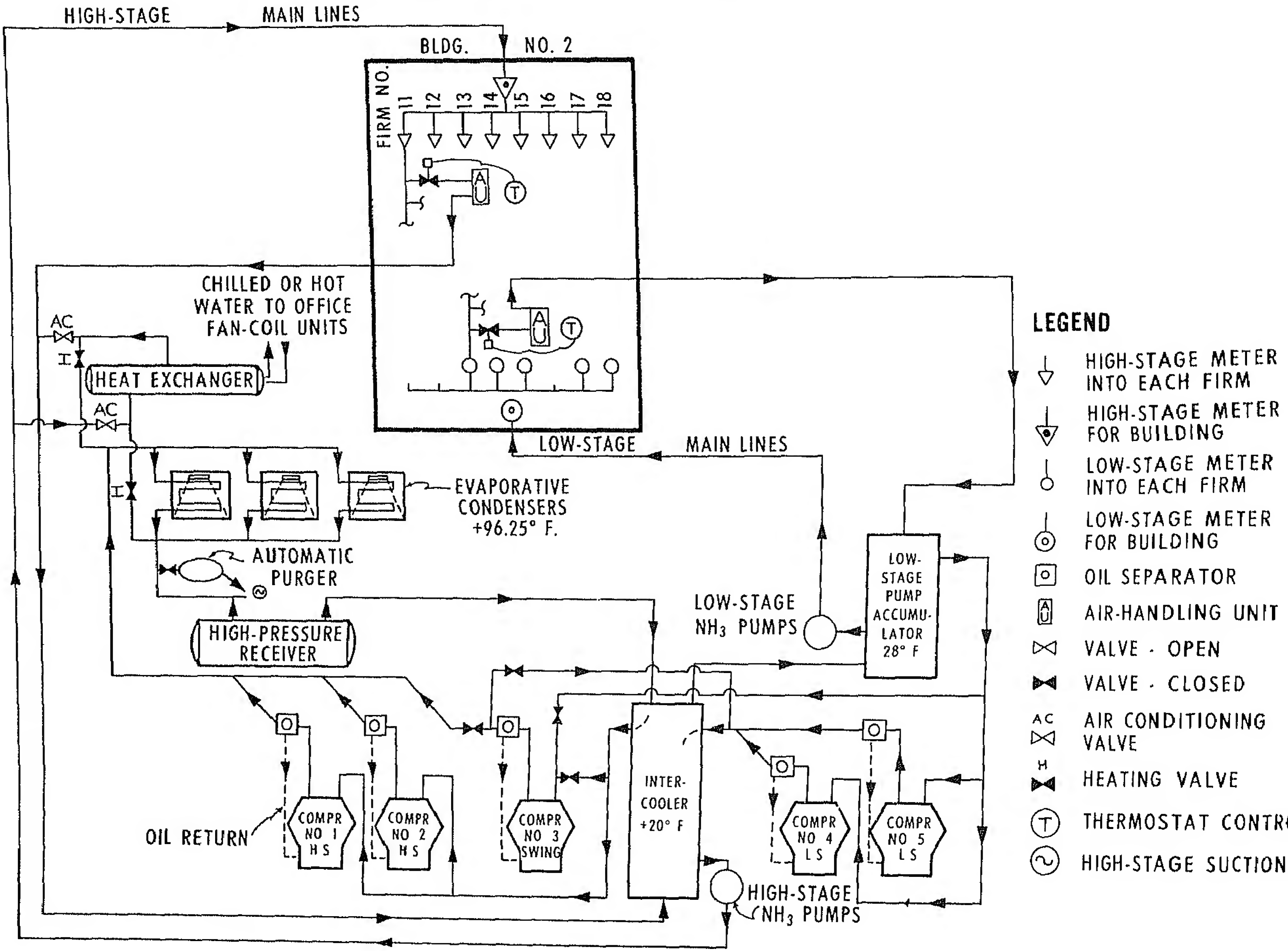


FIGURE 34.—Situation III, Building No. 2, refrigerant-flow diagram.

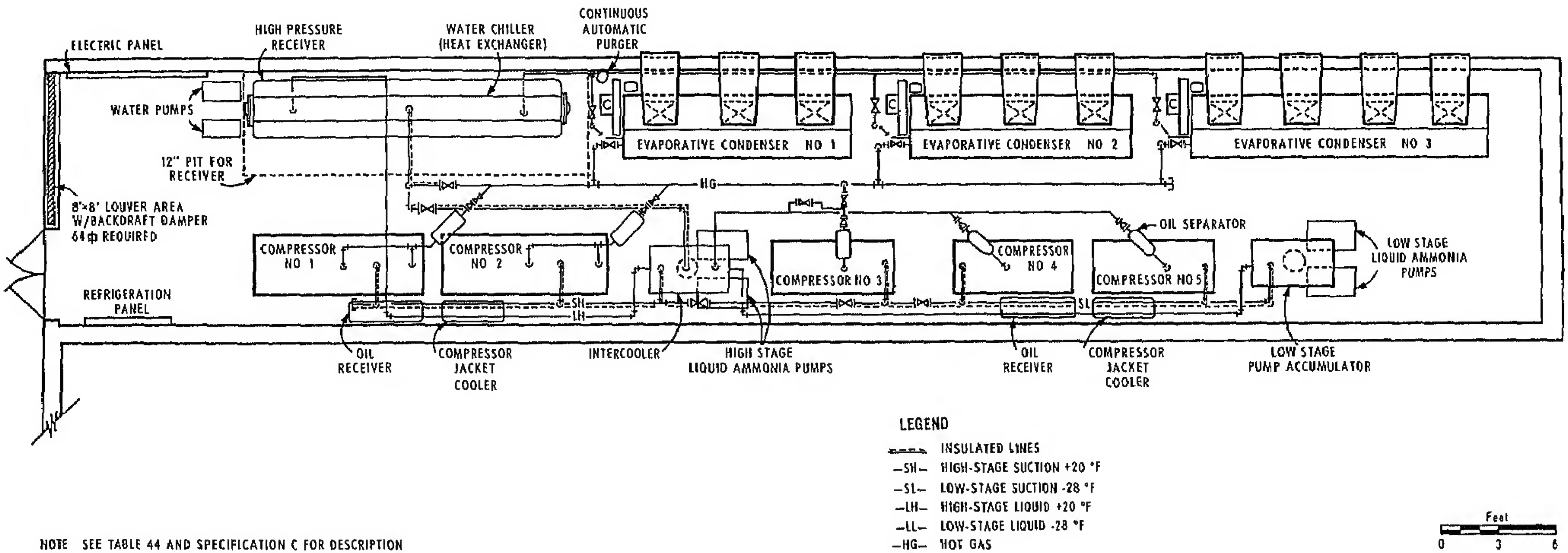
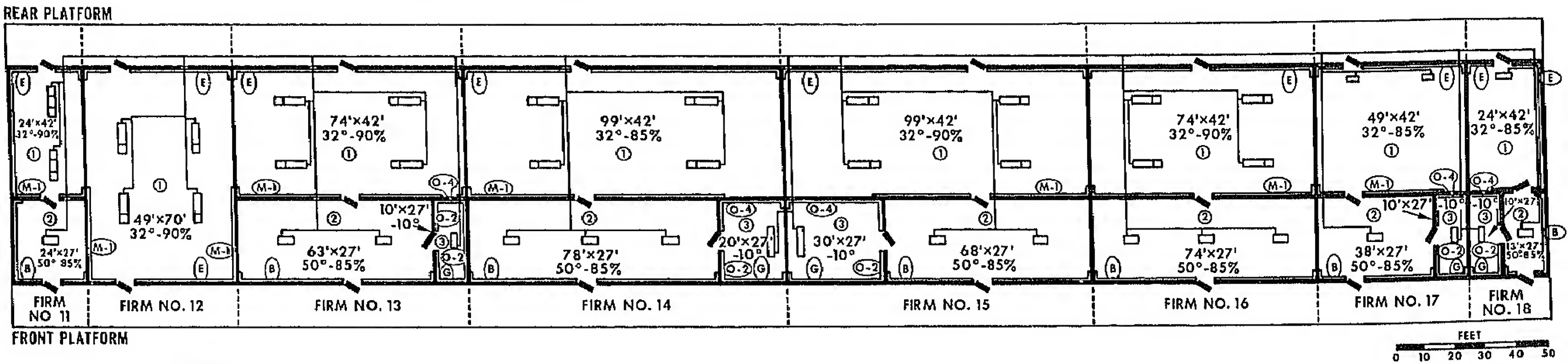


FIGURE 35.—Situation III, Building No. 2, central system equipment room.



LEGEND
▭ ---AIR-HANDLING UNIT

FIGURE 36.—Situation III, Building No. 2 (meat and meat products), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 11 Room 1	Firm 11 Room 2	Firm 12 Room 1	Firm 13 Room 1	Firm 13 Room 2	Firm 13 Room 3	Firm 14 Room 1	Firm 14 Room 2	Firm 14 Room 3
Carcass meat......Pct..	100		75	75			50		
Packaged meat...Pct..			25	25		100	50		100
Refrigeration load B.t.u./hr..	34,879	35,437	93,717	82,835	71,892	74,088	117,442	82,010	138,629
Freezing capacity lb./hr..						430			860

Item	Firm 15 Room 1	Firm 15 Room 2	Firm 15 Room 3	Firm 16 Room 1	Firm 16 Room 2	Firm 17 Room 1	Firm 17 Room 2	Firm 17 Room 3	Firm 18 Room 1	Firm 18 Room 2	Firm 18 Room 3
Carcass meat......Pct..	85			100							
Packaged meat...Pct..	15		100			100		100	100		100
Refrigeration load B.t.u./hr..	114,742	76,380	173,736	82,835	79,893	76,801	41,808	73,569	44,759	28,829	73,519
Freezing capacity lb./hr..			1,200					430			430

EQUIPMENT SCHEDULE

Item	Model							
	AH-3RX	AH-11RX	AH-12RX	AH-13RX	AH-14RX	AH-15RX	AH-16RX	AH-17RX
Air-handling unit:								
Rating...B.t.u. ° F..	2,040	3,140	4,730	7,500	12,500	2,150	2,500	2,850
Air volume....c f m..	2,270	4,950	7,450	11,000	18,300	2,300	2,900	3,700
Fan:								
No. required........	2	2	3	3	5	[1] 1	[1] 1	[1] 1
Size..............in..	14	16	16	18	18			
Fan motor:								
No required........	2	2	3	3	5	1	1	1
Size..............hp..	1/15	1/4	1/4	1/2	1/2	1/4	1/2	1
Type of defrost........	Air	Hot gas	Hot gas	Hot gas	Hot gas	Hot gas	Hot gas	Hot gas
No. of units required ..	13	2	1	3	2	10	4	8

[1] Centrifugal.

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		*° F.*	*In.*		*° F*	*In*	
B......	2 0	Expanded polystyrene.	50	4 0	Expanded polystyrene.	50	1 0	Expanded polystyrene.
E......	3.0	......do......	32	4 0	......do......	32	3.0	Do.
G......	7 0	Fibrous glass.	−10	8.0	Fibrous glass	−10	6 5	Do.
M-1....	2.5	Expanded polystyrene.						
O-2.....	6 5	Fibrous glass.						
O-4.....	5 5	......do..						

NOTES
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3 50° F. rooms are work areas for cutting, boning, packaging, and order assembly operations.
4. Ceiling height is 12 feet in all areas.
5. The "RX" designation in the air-handling model numbers means "recirculated liquid ammonia"

low-stage units should be out of service, the swing unit would help to maintain up to 80 percent of the low-stage capacity required. Duplicate controls are required by compressor No. 3 to convert it into a swing unit; however, this small additional cost gives excellent protection against an emergency breakdown.

The floor plan and air-handling equipment layout are shown in figure 36.

Table 15 contains a summary of all installed costs and owning and operating costs needed to meet the refrigeration requirements for the meat and meat-products dealers.

Other appropriate information used to arrive at these final cost figures is included in the section on costs comparisons. Table 36 shows insulation costs and

TABLE 15.—*Situation III, Building No. 2, installed costs and annual owning and operating costs*

Expenses of refrigeration system[1]	Labor cost	Installed cost including labor
Equipment and insulation costs:	*Dollars*	*Dollars*
Air-handling units	15,100	[2]70,535
Pipe insulation (outside of engine room)	5,320	15,950
Central engine room and piping	7,660	[2]69,170
Pipe and shell insulation (in engine room)	2,350	7,050
Air conditioning (office area)	5,370	15,222
Pipe insulation (air conditioning)	865	2,600
Cold-storage room insulation	33,400	92,215
Cold-storage room doors (per appendix J)		28,000
Refrigerant metering devices		11,731
Total installed cost	70,065	[3]312,473

Annual owning and operating costs:		Total cost
Amortization, capital cost (20 yr @ 6%)	312,473 × 0.08718 =	27,241
Maintenance, insulation (2%/yr.)	120,215 × 0.02 =	2,404
Maintenance, refrigeration, on contract basis	=	9,467
Maintenance, air conditioning, on contract basis	=	672
Insurance, $1.81/thousand adjusted	312,473 × 0.00181 =	566
Taxes, $4.88/thousand adjusted	312,473 × 0.00488 =	1,525
Electric power cost	=	31,500
Total annual owning and operating cost		73,375

[1] Refrigeration load, high-stage 88.6 TR.
Refrigeration load, low-stage 44.4 TR
Air-conditioning load 30.0 TR

[2] Includes proportionate share of interconnecting piping costs.

[3] If office areas are not to be air conditioned, $4,490 can be deducted from this figure, and related owning and operating costs will be lower.

power consumption (kw.hr./yr.) caused by transmission-heat gains. Table 40 is a tabulation of power consumption and electric power costs. Table 44 gives the bill of materials and unit cost.

Building No. 3 (Poultry and eggs)

The refrigerant-flow diagram (fig. 37) and the central engine room illustration (fig. 38) show the operation and components selected to meet the refrigeration requirements of Building No. 3.

Two ammonia pumps and one water pump are provided as standbys to meet any emergency that might arise.

Compressors Nos. 5 and 6, booster-compressors operating in parallel, are 8 percent short of meeting the full load low-stage capacity. However, a diversity factor was not used, which makes this combination acceptable.

Compressors Nos. 1, 2, and 3 handle the high-stage space refrigeration load, the heat rejected by the low-stage units, and the air-conditioning load. These three units provide a capacity slightly in excess of the maximum that could be required at a full load.

Compressor No. 4 is a swing unit capable of operating on either the low or high stage. If one of the three high-stage units should be inoperative, 91 percent of the high-stage capacity could still be maintained. If either of the booster-compressors should fail, 80 percent of the full load low-stage capacity could be maintained. Duplicate controls are required by compressor No. 4 to convert it to a swing unit.

The floor plan and air-handling equipment layout are illustrated in figure 39.

Table 16 contains a summary of all installed costs and owning and operating costs necessary to meet the refrigeration requirements for the poultry and egg dealers.

TABLE 16.—*Situation III, Building No. 3, installed costs and annual owning and operating costs*

Expenses of refrigeration system[1]	Labor cost	Installed cost including labor
Equipment and insulation costs.	*Dollars*	*Dollars*
Air-handling units	17,954	[2]60,375
Pipe insulation (outside of engine room)	5,740	17,200
Central engine room and piping	10,450	[2]70,450
Pipe and shell insulation (in engine room)	2,370	7,100
Air conditioning (office area)	5,370	15,222
Pipe insulation (air conditioning)	865	2,600
Cold-storage room insulation	34,720	89,501
Cold-storage room doors		27,395
Refrigerant metering devices		13,566
Total installed cost	77,469	[3]303,409

Annual owning and operating costs:		Total cost
Amortization, capital cost (20 yr. @ 6%)	303,409 × 0.08718 =	26,451
Maintenance, insulation (2%/yr)	116,890 × 0.02 =	2,338
Maintenance, refrigeration, on contract basis	=	11,650
Maintenance, air conditioning, on contract basis	=	672
Insurance, $1.81/thousand adjusted	303,409 × 0.00181 =	549
Taxes, $4.88/thousand adjusted	303,409 × 0.00488 =	1,481
Electric power cost		27,948
Total annual owning and operating cost		71,089

[1] Refrigeration load, high-stage 66.0 TR.
Refrigeration load, low-stage 48.6 TR.
Air-conditioning load 30.0 TR.

[2] Includes proportionate share of interconnecting piping costs.

[3] If office areas are not to be air conditioned, $7,280 can be deducted from this figure, and related owning and operating costs will be lower.

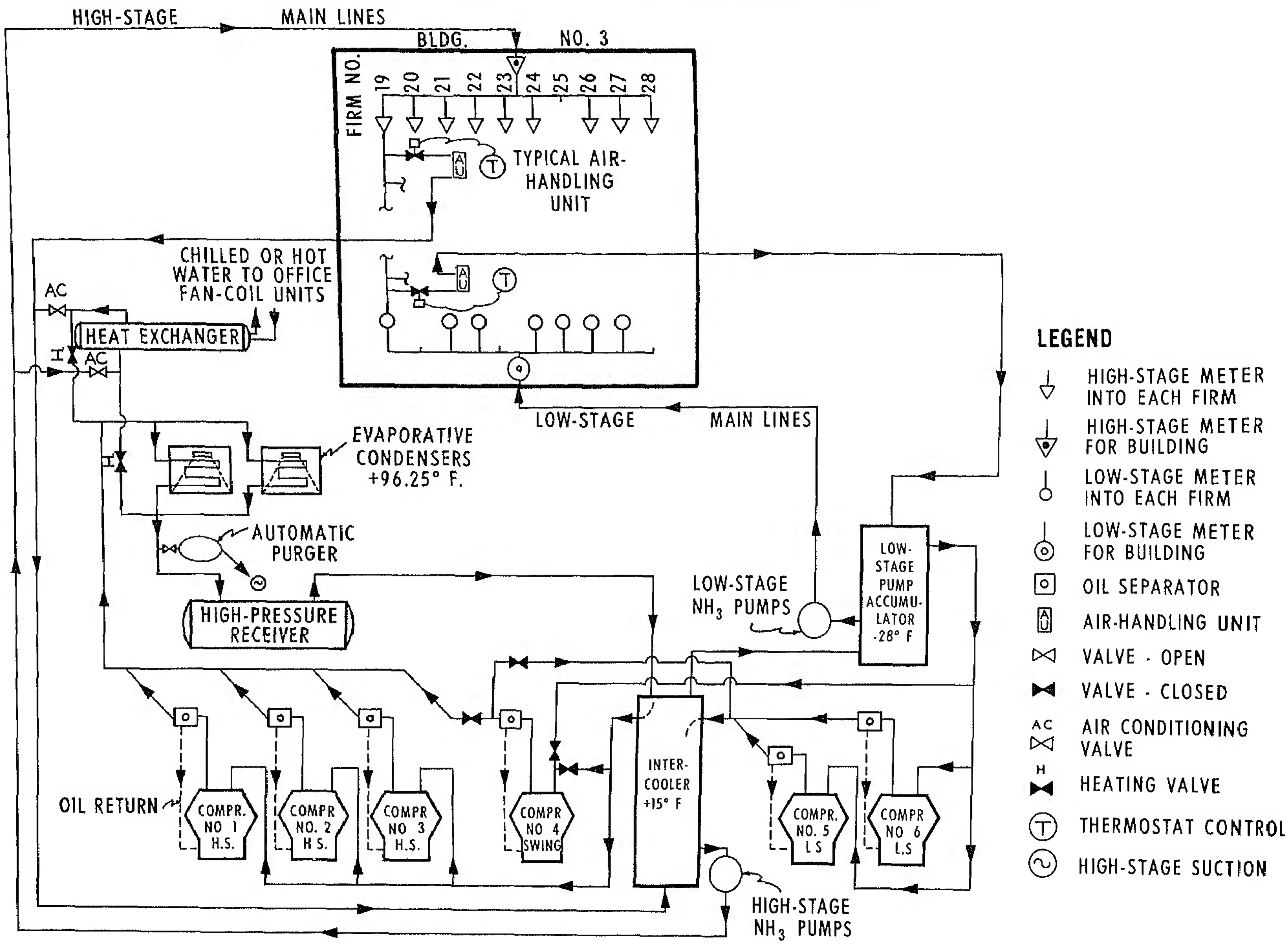


FIGURE 37.—Situation III, Building No. 3, refrigerant-flow diagram.

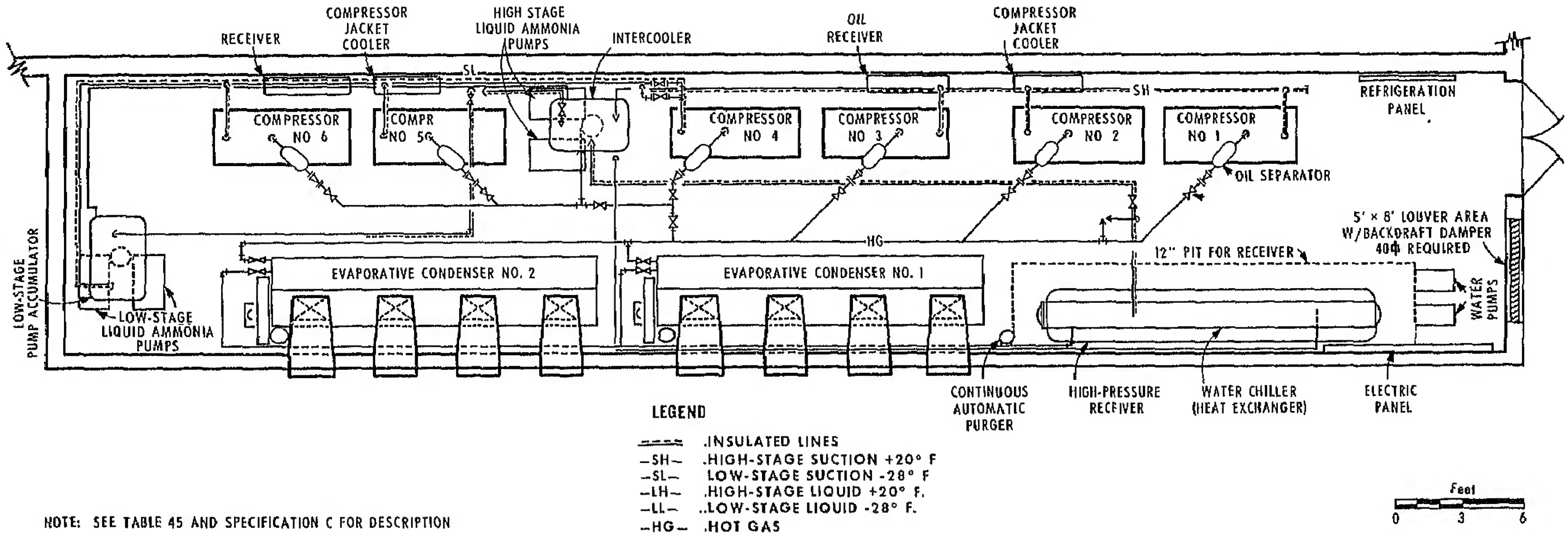


FIGURE 38.—Situation III, Building No. 3, central system equipment room.

Other appropriate information used to arrive at these final cost figures is included in the section on cost comparisons. Table 37 shows, for Situation III, Building No. 3, insulation costs and power consumption (kw.hr./yr.) caused by transmission-heat gains. Table 41 shows power consumption and electric power costs. Table 45 gives the bill of materials and unit cost.

Building No. 4 (Groceries with fresh fruits and vegetables)

The refrigerant-flow diagram (fig. 40) and the central engineroom illustration (fig. 41) show the direct-expansion ammonia system selected to meet the refrigeration requirements of Building No. 4.

Compressor No. 4 is the low-stage booster-compressor. This unit is slightly oversized and will provide an excess capacity of 7.6 percent.

Compressors Nos. 1 and 2 handle the high-stage space refrigeration load, the heat rejected by the low-stage unit, and the air-conditioning load. These two units have an excess capacity of 7.3 percent under full-load conditions.

Compressor No. 3 is a swing unit capable of operating on either the low or high stage. If the booster-compressor should break down, compressor No. 3 can be switched over to maintain 100 percent of the low-stage requirement. If either of the high-stage units should be inoperative, compressor No. 3 can be switched on to help maintain 78 percent of the full-load high-stage capacity. Duplicate controls are required by compressor No. 3 to enable it to operate as either a low- or high-stage unit.

The floor plan and air-handling equipment layout are illustrated in figure 42.

Table 17 contains a summary of all installed costs and owning and operating costs needed to meet the refrigeration requirements for these grocery dealers.

Other appropriate information used to arrive at these final cost figures is included in the section on cost comparisons. Table 38 shows insulation costs and power consumption caused by transmission-heat gains. Table 42 shows power consumption and electric power costs. Table 46 gives the bill of materials and unit cost.

(Text continued on page 60.)

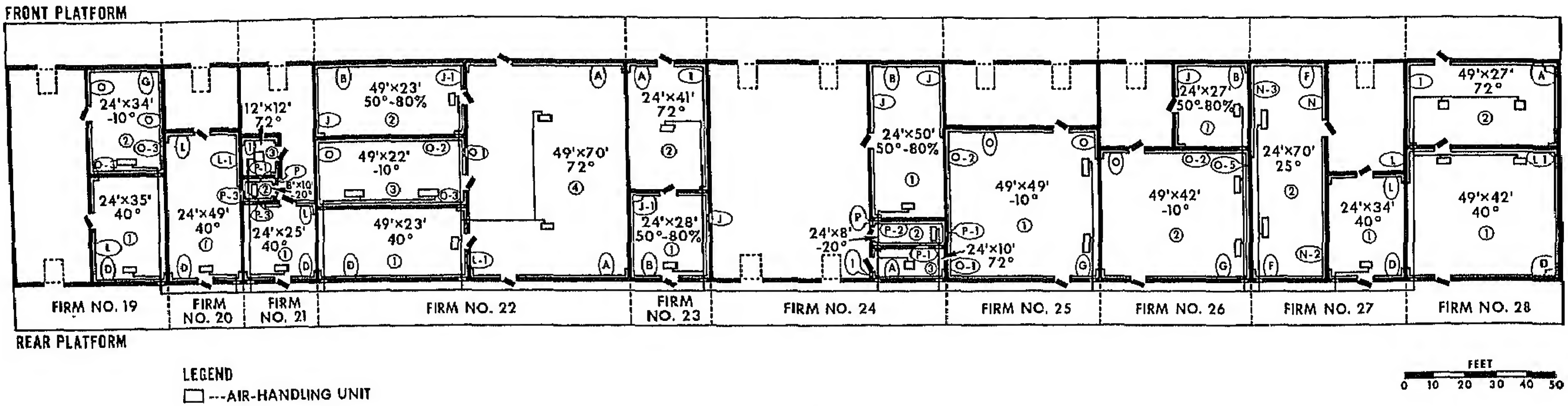


FIGURE 39.—Situation III, Building No. 3 (poultry and eggs), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 19		Firm 20	Firm 21			Firm 22				Firm 23		Firm 24			Firm 25	Firm 26		Firm 27		Firm 28	
	Room 1	Room 2	Room 1	Room 1	Room 2	Room 3	Room 1	Room 2	Room 3	Room 4	Room 1	Room 2	Room 1	Room 2	Room 3	Room 1	Room 1	Room 2	Room 1	Room 2	Room 1	Room
Refrigeration load B t.u /hr..	41,340	163,475	56,782	32,696	15,493	16,268	57,378	67,247	201,049	80,265	39,760	34,905	60,090	33,969	9,256	94,059	42,334	75,093	42,800	82,535	80,271	46,131
Room usage or product handled....	Poultry	Poultry	Poultry	Poultry	Poultry	Cutting up poultry.	Poultry	Shell eggs.	Poultry	Order assembly.	Shell eggs	Egg grading, order assembly.	Shell eggs.	Egg freezing	Egg breaking.	Poultry	Shell eggs.	Frozen eggs.	Poultry	Crusted chicken.	Poultry	Order assembly
Freezing capacity lb./hr.		958			38				1,200					141 4						208		

EQUIPMENT SCHEDULE

Item	Model							
	AH-2RX	AH-3RX	AH 6RX	AH-7RX	AH-11RX	AH-12RX	AH-13RX	AH-14RX
Air-handling unit:								
Rating B.t.u./hr./° F..	1,430	2,040	4,330	7,500	3,140	4,730	7,500	12,500
Air volume....c.f.m..	1,515	2,270	6,500	10,500	4,950	7,450	11,000	11,000
Fan:								
No. required.......	1	2	3	3	2	3	3	3
Size............in..	16	14	16	18	16	16	18	18
Fan motor:								
No. required.......	1	2	3	3	2	3	3	3
Size...........hp..	1/15	1/15	1/4	1/2	1/4	1/4	1/2	1/2
Type of defrost.......	Air	Air	Air	Air	Hot gas	Hot gas	Hot gas	Hot gas
No. of units required...	4	3	4	7	3	2	5	1

NOTES:
1. Insulation thicknesses have not been subtracted from dimensions shown.
2. Humidities shown are minimum requirements.
3. Ceiling height is 20 feet in all areas except in firms 21-2, 21-3, 24 2, and 24-3, where height is 10 feet.
4. The "RX" designation in the air-handling model numbers means "recirculated liquid ammonia."

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		*° F.*	*In.*		*° F.*	*In.*	
A......	1.0	Expanded polystyrene.	72	3.0	Expanded polystyrene.	72	....	None
B......	2 0	......do......	50	4.0	......do......	50	1.0	Expanded polystyrene
D......	2 5	......do......	40	4.0	......do......	40	2.5	Do.
F......	3.5	......do......	25	4.5	......do......	25	3.5	Do.
G......	7 0	Fibrous glass.	−10	8.0	Fibrous glass.	−10	6 5	Do.
I......	1 5	Expanded polystyrene.	−20	8.5	......do......	−20	7.0	Do.
J......	3 5	......do......						
J-1......	3.0	......do......						
L......	4.0	......do......						
L-1......	3.5	......do......						
N......	4 5	......do......						
N-2......	2.0	......do......						
N-3......	2.0	......do......						
O......	8.5	Fibrous glass.						
O-1......	8.0	......do......						
O-2......	6 5	......do......						
O-3......	6 0	......do......						
O-5......	5.5	......do......						
P......	9.0	......do......						
P-1......	8.5	......do......						
P-2......	7.0	......do......						
P-3......	6.5	......do......						

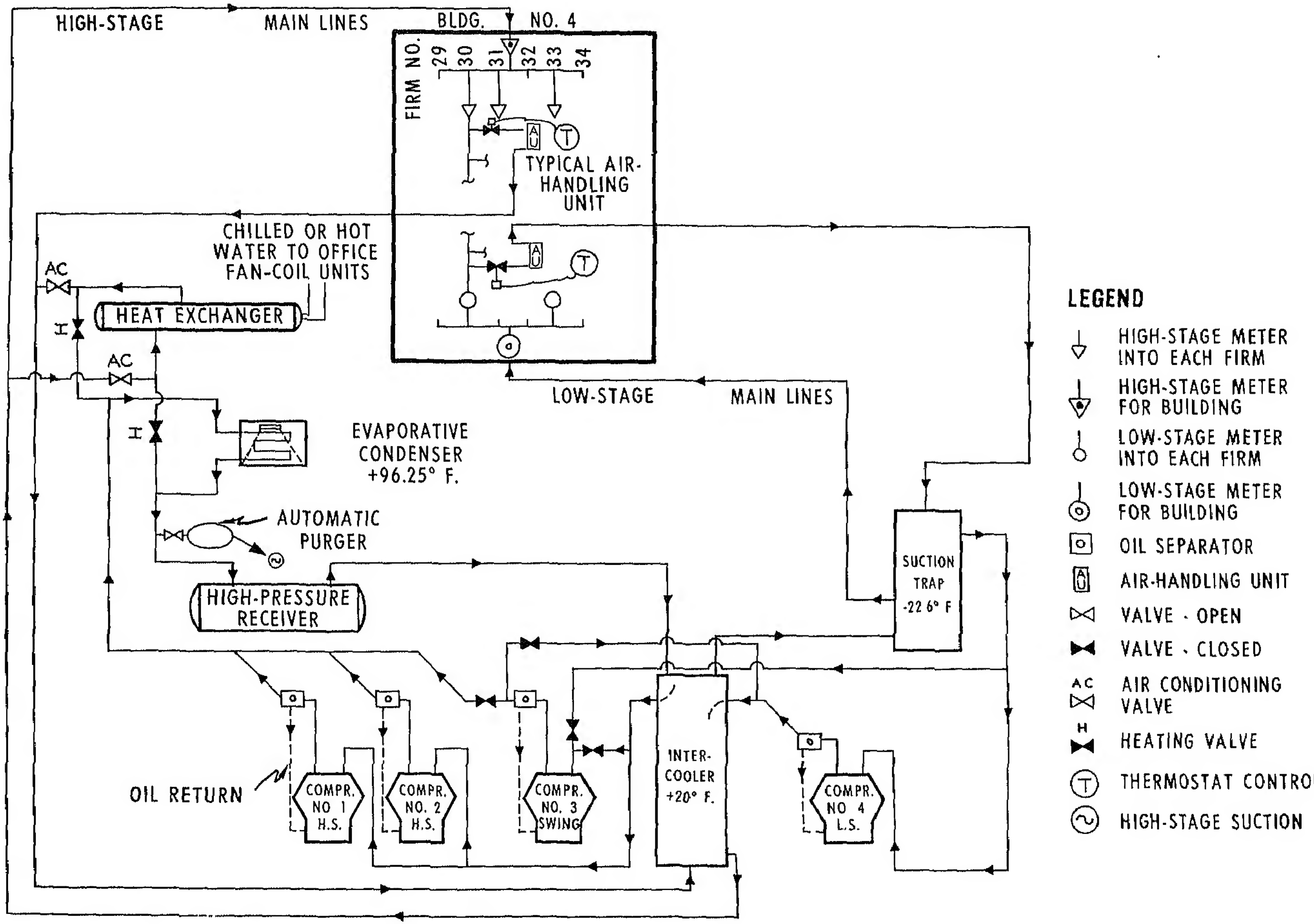


FIGURE 40.—Situation III, Building No. 4, refrigerant-flow diagram.

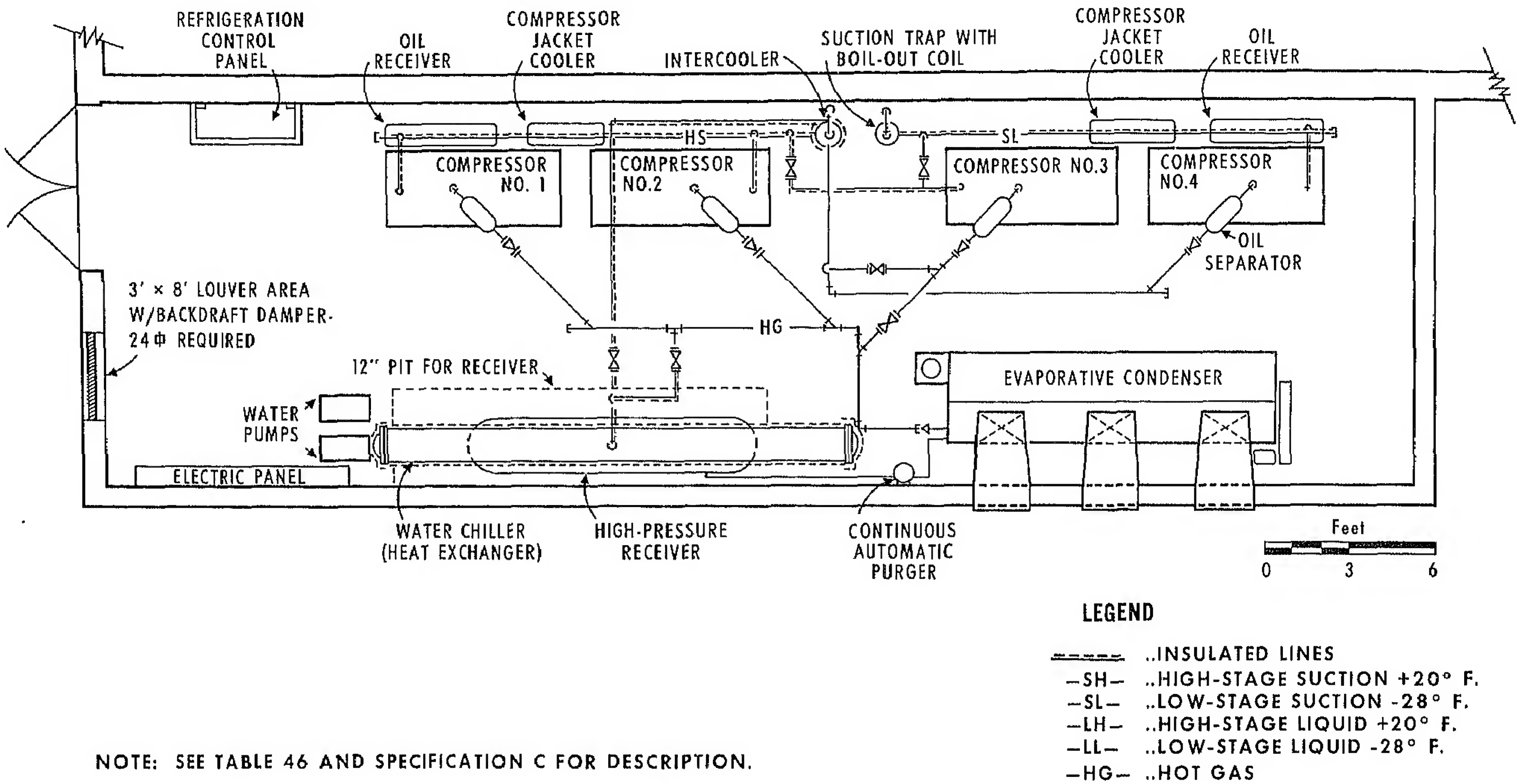
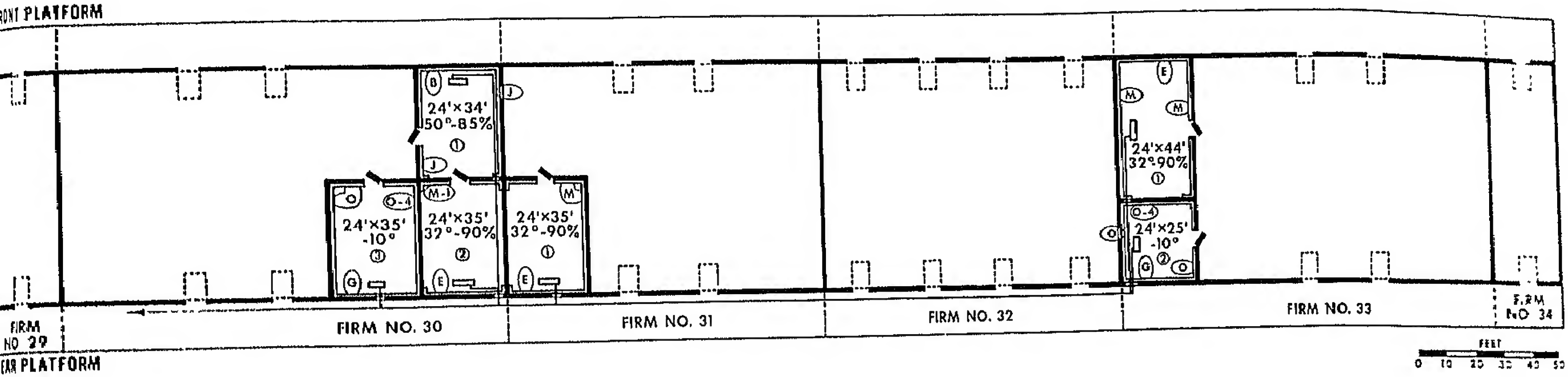


FIGURE 41.—Situation III, Building No. 4, central system equipment room.



LEGEND
[ ]---AIR-HANDLING UNIT

FIGURE 42.—Situation III, Building No. 4 (groceries with fresh fruits and vegetables), floor plan and air-handling equipment layout.

REFRIGERATION SCHEDULE

Item	Firm 30			Firm 31	Firm 33	
	Room 1	Room 2	Room 3	Room 1	Room 1	Room 2
Refrigeration load B.t.u./hr..	76,305	80,223	50,255	83,806	92,865	30,935

EQUIPMENT SCHEDULE

Item	Model			
	AH-7DX	AH-12DX	AH-13DX	AH-14DX
Air-handling unit:				
Rating..........B.t.u./° F..	6,000	3,780	6,000	10,000
Air volume..........c.f.m..	10,500	7,450	11,000	18,300
Fan:				
No. required..........	3	3	3	5
Size..........in..	18	16	18	18
Fan motor:				
No. required..........	3	3	3	5
Size..........hp..	1/2	1/4	1/2	1/2
Type of defrost..........	Air	Hot gas	Hot gas	Hot gas
No. of units required..........	1	1	1	3

INSULATION SCHEDULE

Code	Wall material		Ceiling material			Floor material		
	Thickness	Type	Room temperature	Thickness	Type	Room temperature	Thickness	Type
	In.		*° F.*	*In.*		*° F.*	*In.*	
B......	2.0	Expanded polystyrene.	50	4.0	Expanded polystyrene.	50	1.0	Expanded polystyrene.
E......	3.0	......do......	32	4.0	......do......	32	3.0	Do.
G......	7.0	Fibrous glass.	−10	8.0	Fibrous glass	−10	6.5	Do.
J......	3.5	Expanded polystyrene.						
M......	4.5	......do......						
M-1......	2.5	......do......						
O......	8.5	Fibrous glass......						
O-4......	5.5	......do......						

NOTES.
1. Insulation t[illegible] ...tions shown
2. Humidities
3. Ceiling heig[illegible]
4. The "DX" [illegible] direct expansion ammonia."

TABLE 17.—*Situation III, Building No. 4, installed costs and annual owning and operating costs*

Expenses of refrigeration system[1]	Labor cost	Installed cost including labor
Equipment and insulation costs:	*Dollars*	*Dollars*
Air-handling units	3,720	[2]17,160
Pipe insulation (outside of engine room)	2,690	8,000
Central engine room and piping	4,870	[2]33,555
Pipe and shell insulation (in engine room)	1,160	3,350
Air conditioning (office area)	5,370	15,222
Pipe insulation (air conditioning)	865	2,600
Cold-storage room insulation	9,620	25,651
Cold-storage room doors		5,890
Refrigerant metering devices		3,825
Total installed cost	28,295	[3]115,253

Annual owning and operating costs:		Total cost
Amortization, capital cost (20 yr. @ 6%)	115,253 × 0.08718 =	10,048
Maintenance, insulation (2%/yr.)	31,541 × 0.02 =	631
Maintenance, refrigeration, on contract basis	=	5,072
Maintenance, air conditioning, on contract basis	=	672
Insurance, $1.81/thousand adjusted	115,253 × 0.00181 =	209
Taxes, $4.88/thousand adjusted	115,253 × 0.00488 =	562
Electric power cost	=	15,264
Total annual owning and operating cost		32,458

[1] Refrigeration load, high-stage 27.7 TR.
Refrigeration load, low-stage 7.2 TR.
Air-conditioning load 30.0 TR.
[2] Includes proportionate share of interconnecting piping costs.
[3] If office areas are not to be air conditioned, $7,720 can be deducted from this figure, and related owning and operating costs will be lower.

Summary of all Costs

The total installed cost for the refrigeration equipment and insulation (including refrigeration doors) for all four buildings in Situation III is $966,661. The total annual owning and operating cost for Situation III is $239,463 (table 18).

TABLE 18.—*Situation III, summary of costs for all buildings*

Building No.	Installed cost	Annual owning and operating cost
	Dollars	*Dollars*
1	235,526	62,541
2	312,473	73,375
3	303,409	71,089
4	115,253	32,458
Total	966,661	239,463

AIR CONDITIONING-HEATING FOR SITUATIONS II AND III

Heating is, of course, a must. In new buildings, air conditioning is also considered a necessity, not a luxury, to improve both the efficiency and the morale of personnel.

In Situation I, where unitary package systems are used by each firm, the air conditioning is omitted, because completely separate units would be required. If a food distribution center uses package refrigeration systems similar to those in this situation, separate air-conditioning units can be selected and the cost added to that established for the refrigeration units. A separate heating system would be required unless it was combined with the air conditioning.

When a central system is used, as in Situations II and III, the air-conditioning and heating requirements can be handled by the central refrigeration equipment at a nominal first-cost addition. Air-handling units of some kind must be supplied in each office space, but the heating or cooling medium for these units is supplied from the central system equipment room or building.

The architectural detail for an office area can vary greatly with the materials used, building orientation, and geographic location. All affect the air-conditioning or heating load on which equipment selections are based. To present a broad view of what would be required to provide conditioned air to the office areas of a food distribution center, this section of the report outlines average design data and conservative equipment selections.

The following design criteria are assumed:

Summer temperature	95° F. outside,	78° F. inside, dry bulb, or
	76° F. outside,	65° F. inside, wet bulb
Winter temperature	0° F. outside,	70° F inside, dry bulb.
Square feet of floor area/ton of refrigeration		200 ft.²/TR.
Water circulation		3 g.p.m./TR.
Water temperature range		8° F.
Water temperature from heat exchanger, summer		40° F
Water temperature from heat exchanger, winter		105° F

The design criteria listed above are average values used to make equipment selections and approximate load calculations.

To set up the air-conditioning and heating zones, each 25-foot bay is divided into two 12-foot, 6-inch zones with individual units and controls.

System Selection and Operation

In determining the type of system to be used, flexibility, quality, and owning-operating costs, as well as physical design data, are important considerations.

A chilled- or hot-water recirculation system best meets these criteria. The conditioned water is circulated by pumps from the central-system equipment room or building to the individual office areas, where some type of air-handling equipment is used to condition the space. For this study, one free-standing, fan-coil console unit is installed per zone. If preferred, an individual firm could use one air-handling unit with ductwork to handle its entire office space; or it might choose to reduce the two 12-foot, 6-inch zones to three 8-foot, 4-inch zones, which would require three fan-coil units instead of two.

Selection of the fan-coil units is based on essentially equal heat losses or gains at the design conditions. Should an architectural designer wish to use the specific selections listed in table 19, the heat transfer at peak summer and winter conditions through the walls, floor, ceiling, and windows should be equal, and should total approximately 11,340 B.t.u./hr. per 25-foot module.

Hot or cold water is supplied to the individual fan-coil units from the central-system equipment room or building. A shell-and-tube heat exchanger is incorporated in the refrigeration system to take advantage of waste heat from the refrigerated spaces. See figure 24, the refrigerant flow diagram for Situation II. In the summer, when cooling is required, the heat exchanger (water chiller) is piped into the low side of the system, where it acts as an evaporator. Recirculated water from the office units passes through the tubes, where it is cooled from approximately 48° to 40° F. In the winter, or when heating is desired, a manual changeover of the valves places the heat exchanger on the high side of the refrigeration system where, acting as a condenser, it produces hot water.

In Chicago, as in most of the major population centers of the United States, the heating season is of longer duration than the cooling season. In such areas, the refrigeration provides a major source of waste heat that is normally thrown to the outside. Transferring this waste heat to the office areas is an economical way of heating.

Separate compressors are not used for the air-conditioning duty. On central systems of the sizes that are used in this study, the air-conditioning load is such a small part of the total refrigeration load that it is economically feasible to use the same compressors, condensers, and other components to handle everything.

Refer to figures 31, 34, 37, and 40, for illustrations showing the heat-exchanger operation applications.

Two water pumps are included in each central-system equipment room or building, to circulate the hot or cold water to the office fan-coil units. One pump is a standby. The pumps are illustrated both in the refrigerant-flow diagrams and in the figures illustrating the equipment-building layouts.

Equipment Cost and Bill of Materials

The costs for furnishing and installing the heat exchangers (water chillers) and water pumps are included in the various bills of materials for the central-system equipment rooms and building. Table 19 is a summary of these costs.

Each building has a 30-ton air-conditioning load, which makes the cost differential between situations so small that the costs can be considered the same. For practical purposes, a cost estimate of $661/TR is established for air conditioning.

TABLE 19.—*Air-conditioning and heating bill of materials and costs*

Situation and quantity	Equipment description	Unit cost
		Dollars
Situation II.		
160	Fan-coil units, free-standing, vertical, per specifications and including controls, valves, piping, insulation, and installation	425
1	Horizontal shell-and-tube heat exchanger, 30″ diameter × 16′ long, complete with automatic valves and float-switch level control, per specifications	9,430
2	Water pumps, per specifications	1,350
	Total cost—Situation II	80,130
Situation III:		
160	Fan-coil units, same as described above	425
3	Horizontal shell-and-tube heat exchangers, 14″ diameter × 16′ long, described as above (for Buildings Nos. 1, 2, and 3)	1,990
1	Horizontal shell-and-tube heat exchanger, 14″ diameter × 16′ long, described as above, (for Building No. 4)	1,800
8	Water pumps, per specifications	400
	Total cost—Situation III	78,970

Operating Costs

The operating costs for air conditioning are determined by the final architectural design, but they would accrue at $0.324/TR/day in Situation II, and at $0.488/TR/day in Situation III.

Operating costs for heating consist only of the cost of operating the water pump plus the cost for a slight additional kw.-hr./TR on the compressors. The heat used to raise the water temperature is waste heat, which ordinarily would be dissipated to the outside air through the evaporative condensers.

The owning and operating costs are included with the refrigeration equipment. See tables 13 through 17 for a summary of each situation and building.

Deductions for Omitting Air Conditioning

If air conditioning is to be omitted in the office areas, the following deductio can be made for each central-system equipment room or building:

Situation II (table 13)	$18,620
Situation III, Building No. 1 (table 14)	5,180
Situation III, Building No. 2 (table 15)	4,400
Situation III, Building No. 3 (table 16)	7,280
Situation III, Building No. 4 (table 17)	7,720

These deductions result from different refrigeration components that wou be selected if the air-conditioning load of 30 tons per building were omitted. Smal compressors and condensers would be used, and installation time would be le There is no deduction for the room fan-coil units, because they would still be quired to supply heating.

COST COMPARISONS FOR THE THREE SITUATIONS

General Comparison of Total Costs

Table 20 shows individual costs for Situations I, II, and III. By comparing these costs, it can be seen that Situation II (one central refrigeration system for all four buildings) costs less to install and less to own and operate than either Situation I (each firm providing its own individual refrigeration system) or Situation III (each building having a separate central refrigeration system). In total capital expenditures, which include all costs associated with furnishing and installing the insulation and refrigeration equipment, Situation I costs 4.3 percent more than Situation II, and in annual cost of owning and operating, it costs 61.9 percent more than Situation II. Situation III costs 8.2 percent more than Situation II for the insulation and equipment, and 25.4 percent more to own and operate.

Two major items contributing to these cost differences are insulation and electric power costs. Table 20 compares costs on $/ton of refrigeration, $/square foot of floor space, and $/cubic foot of refrigerated area. The square-foot and cubic-foot values are based on refrigerated spaces only.

The following results are specifically pointed out in this report:

1. The insulation cost can be substantially reduced by using one central system. The transmission-heat gains are not as critical to a large central system as they are to package systems.

2. Air conditioning and heating for the office areas can be handled by part of the central-system refrigeration equipment at relatively low first costs. Situation I's package systems would require completely separate units to handle the air conditioning.

3. The electric power costs and owning and operating costs are substantially reduced when one central system is employed, even though they include costs for heating and air conditioning.

TABLE 20.—*Costs summary for all situations*

Individual costs[1]	Situation No. I	II	III
	Dollars	*Dollars*	*Dollars*
Refrigeration equipment	535,676	[2]540,003	[2]606,390
$/TR	1,255	[2]943	[2]1,003
$/ft.²	6.22	5.33	5.93
$/ft.³	.382	.328	.361
Insulation	395,922	344,274	360,271
$/TR	928	745	860
$/ft.²	4.60	4.00	4.19
$/ft.³	.282	.246	.257
Total capital expenditures	931,598	[2]893,277	[2]966,661
$/TR	2,183		
$/ft.²	10.82	9.33	10.12
$/ft.³	.664	.574	.621
Electric power costs/yr.	134,137	[2]77,016	[2]104,070
$/TR/24 hr.	.873	.463	.612
Owning and operating/yr.	309,157	190,941	230,463
$/TR/yr.	724	389	565
$/ft.²/yr.	3.59	2.00	2.64
$/ft.³/yr.	.22	.128	.162

[1] Based on refrigerated space only.
[2] Includes cost figures for 120 tons of air-conditioning/heating duty as a part of the centr refrigeration system.

4. The total capital expenditures for one central system are substantially less han for package systems, even though air conditioning and heating are included. The smaller the central system, however, the closer its initial cost approaches those of package systems.

For a comparison of the breakdown costs for such items as maintenance, amortization, insurance, and taxes, refer to the individual cost summaries of each situation.

Cost Calculations for Insulation and Electric Power—all Situations

Tables 22 through 46 include all figures used in determining the final insulation costs and operating costs. The calculations for Situation I are illustrated for one sample firm. Complete tables are included for situations II and III.

The installed cost per square foot for each type of insulation has been shown in tables 3, 4, 5, 6, 7, and 8. The cost per square foot is multiplied by the number of square feet installed to derive the total insulation cost per building and situation. Table 22 illustrates the manner in which the calculations were made for Situation I. Tables 28 through 31 and 35 through 38 list these calculations for Situations II and III.

The electrical costs are based on rates of the Commonwealth Edison Company in Chicago, Illinois, as of 1965 (table 21).

TABLE 21.—*Electricity costs by Commonwealth Edison Company in Chicago, Ill., in 1965*

Amount used	Rate
Demand cost:[1]	
First 200 kw	\$2.00/kw.
Next 800 kw	1.80/kw.
Next 2,500 kw	1.65/kw.
Next 11,500 kw	1.45/kw.
Next 85,000 kw	1.35/kw.
Power cost:[2]	
10 kw.-hr	\$1.40 flat charge
490 kw.-hr	2.8¢/kw.-hr.
2,000 kw.-hr	2.4¢/kw.-hr.
3,500 kw.-hr	1.75¢/kw.-hr.
24,000 kw.-hr	1.35¢/kw.-hr.
70,000 kw.-hr	1.00¢/kw.-hr.
400,000 kw.-hr	.75¢/kw.-hr.[3]
above 500,000 kw.-hr	.65¢/kw.-hr.[3]

[1] Based on maximum electric power that must be available.
[2] Based on electric power actually used.
[3] When the energy consumption exceeds 450 times the demand, this rate becomes 0.60¢/kw.-hr.

In calculating the electrical costs for all situations, the following factors were used:

Lights and occupancy	12 hr./day
Product loading on annual basis	60 percent of capacity
Miscellaneous motors, (cutting rooms, etc.)	12 hr./day
Battery chargers, truck refrigeration, etc	Not considered

The demand cost is based on the maximum electric power that the electric company must have available at all times. The kilowatt rating of all lights and of fan, compressor, and pump motors, plus any miscellaneous electrical equipment that can be operating simultaneously are totaled to determine this maximum power demand. To determine the amount of demand cost, the total kilowatt figure is applied to the appropriate cost in table 21. Table 22 illustrates the manner in which the demand costs were calculated for the individual firms in Situation I. Tables 32 and 39 through 42 list the demand cost calculations for Situations II and III.

The energy cost is based on the kilowatts of electric power actually used by a customer in operating his refrigeration equipment, lights, and other miscellaneous electrical equipment. To forecast this usage accurately on a monthly basis, it is necessary to convert the heat removed from the refrigerated spaces (B.t.u./hr.) to electric power consumed (kw.-hr./month). This figure is equivalent to the power required by the refrigeration equipment. The electricity required for the lights and other miscellaneous equipment can be taken as fixed amounts based on operating hours and size of facilities.

Tables 28 through 31 and 35 through 38 include all calculations of power consumed (kw.-hr./yr.) due to heat gains through the floor, ceiling, and walls.

Example:

kw.-hr./yr. = Q/A × Area × Hr. × Conversion factor

where.

Q/A (B.t.u./hr.–ft.^{2}) = the heat gain though a surface. This is listed in tables 3, 4, 5, 6, 7, and 8 for the various types of insulation.

Area (ft.^{2}) = the surface area of the floor, ceiling, or walls.

Hours (hr./yr.) = the number of hours for which the specific temperature difference exists across that surface. For outside walls, this is based on the weighted temperature difference.

Conversion factor (kw./B.t.u./hr. × 10^{-3}) =

$$\frac{\text{Horsepower} \times 0.746 \text{ kilowatts/horsepower}}{\text{Tons refrigeration} \times \text{efficiency} \times 12{,}000 \text{ B.t.u./hr.–TR}}$$

The calculation for the conversion factor includes the horsepower of all the refrigeration equipment at an 80-percent efficiency. For refrigerated rooms of 32° F. and higher, the conversion factor works out to 0.109 kw./B.t.u./hr. × 10^{-3}, and for freezers the value is 0.214 kw./B.t.u./hr. × 10^{-3}. These values vary slightly in Situations II and III, but the difference is very slight because the hp./TR remains fairly constant. Thus, the same factors were used for both situations.

For the firms in Situation I, this conversion factor varies for each room. Calculate by adding the operating kw. of the condenser units to the operating kw. of the air-handling units and dividing by the refrigeration load (B.t.u./hr.) The power consumed (kw.–hr./yr.) to handle the heat gains through air changes, product loads, and miscellaneous loads is determined by:

kw.–hr./yr. = Q × hrs. of operation × conversion factoı

where:

Q (B.t.u./hr.) = heat gain.

hr. of operation (hr /yr.) = the number of hours these loads exist.

Conversion factor = same as developed above.

The power consumed (kw.–hr./yr.) to handle the electric lights and miscellaneous motors is found by:

$$\text{kw.–hr./yr.} = \frac{\text{Electrical rating (watts)} \times \text{hr. of operation}}{\text{1,000 watts/kilowatt}}$$

(Text continued on pag

TABLE 22.—*Situation I, Firm No. 25-1, sample calculations of insulation costs and electric power consumption and costs*

Insulation code[1] and heat source	Q/A[2]	Area	Hours/year	Conversion	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
	*B.t.u./hr.-ft.*²	*Ft.*²	*Hours*	*Kw./B.t.u./hr. × 10⁻³*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
O	1.474	980	8,760	0.284	3,597	2.015	1,974
G	1.194	980	8,760	.284	2,918	1.852	1,818
O	1.474	160	8,760	.284	680	2.015	322
O-2	1.169	980	8,760	.284	2,849	1.852	1,818
Roof	1.358	2,401	8,760	.284	8,053	1.982	4,760
Floor	1.49	2,401	8,760	.284	8,928	1.047	2,510
						Total cost	13,202
	Q						
	B.t.u./hr.						
Lights	8,160		4,380	.284	10,176		
Motors	6,225		8,760	.284	15,454		
People	1,400		4,380	.284	1,746		
Air	31,700		5,256	.284	47,314		
Product	24,900		5,256	.284	37,176		
	Watts						
Lights	5,000		4,380		21,900		
Motors	2,238		8,760		19,605		

Total annual....180,396 kw.–hr./yr. ÷ 12 mo./yr. = 15,033 kw.–hrs./mo.

Demand................35.6 kw.–hr. × 2.00 = $ 71.20

Energy................6000 kw.–hr. × .0207 = 124.35

9033 kw.–hr. × .0135 = 121.95

Average power cost............$317.50/month

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.

[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 23.—*Situation I, bill of materials and unit costs for refrigeration equipment*

TABLE 23.—*Situation I, bill of materials and unit costs for refrigeration equipment*—Continued

Firm number and proposal	Room temperature	Air unit model number	Number required	Installed cost	Condensing unit model number	Number required	Installed cost[1]
	°F.		Number	Dollars		Number	Dollars
1: Base	32	AH-4	1	845	CU 2-12	2	2,071
Alternate	32	AH-4	1	845	CU 5-12	1	1,571
2: Base	40	AH-4	1	880	CU ¾-12	2	[2]1,716
Alternate	40	AH-4	1	880	CU 1-12	1	[2]1,146
3: Base	45	AH-7	3	4,900	CU 15-12	2	5,105
Alternate	45	AH-7	3	4,900	CU 10-12	2	4,315
4: Base	32	AH-10	3	7,900	CU 15-22	2	6,640
Alternate	32	AH-10	3	7,900	CU 15-22	2	6,040
5: Base	50	AH-7	2	3,540	CU 10-12	2	4,326
	32	AH-10	2	5,410	CU 7½-12	2	4,126
Alternate	50	AH-7	2	3,540	CU 15-12	1	2,980
	32	AH-10	2	5,410	CU 15-22	1	3,810
6: Base	50	AH-7	4	5,800	CU 15-12	3	8,007
Alternate	50	AH-7	4	5,800	CU 15-12	3	8,007
7: Base	50	AH-7	2	3,925	CU 15-12	2	4,981
	50	AH-7	2	3,925	CU 10-12	2	4,500
	32	AH-10	2	6,200	CU 7½-12	2	4,300
Alternate	50	AH-7	2	3,925	CU 15-22	2	7,310
	50	AH-7	2	3,925			
	32	AH-10	2	6,200	CU 15-22	1	4,040
8: Base	50	AH-7	1	1,790	CU 7½-12	2	3,620
	32	AH-10	2	5,880	CU 10-12	2	4,240
Alternate	50	AH-7	1	1,790	CU 10-22	1	2,160
	32	AH-10	2	5,880	CU 15-22	1	3,720
9: Base	50	AH-7	6	9,775	CU 20-22	4	12,575
Alternate	50	AH-7	6	9,775	CU 20-22	4	12,575
10: Base	32	AH-9	2	3,480	CU 10-12	2	4,519
	50	AH-7	2	3,410	CU 10-12	2	4,281
Alternate	32	AH-9	2	3,480	CU 15-12	1	3,590
	50	AH-7	2	3,410	CU 15-12	1	3,220
11: Base	32	AH-8	2	2,940	CU 5-12	2	3,290
	50	AH-2	2	1,495	CU 3-12	2	2,577
Alternate	32	AH-8	2	2,940	CU 7½-12	1	2,480
	50	AH-2	2	1,495	CU 7½-12	1	1,990
12: Base	32	AH-9	3	5,250	CU 10-12	2	4,250
Alternate	32	AH-9	3	5,250	CU 7½-12	2	3,910
13: Base	32	AH-9	3	5,350	CU 7½-12	2	4,460
	50	AH-3	3	2,670	CU 7½-12	2	3,600
	−10	AH-10	1	2,760	CU 10-22	3	7,791
Alternate	32	AH-9	3	5,350	CU 10-12	1	3,470
	50	AH-3	3	2,670	CU 10-12	1	2,520
	−10	AH-10	1	2,760	CU 7½-502	3	7,000
14: Base	32	AH-9	4	7,240	CU 7½-12	2	4,086
	50	AH-3	3	2,760	CU 7½-12	2	3,360
	−10	AH-10	2	5,560	CU 10-22	5	13,200
Alternate	32	AH-9	4	7,240	CU 15-22	1	3,900
	50	AH-3	3	2,760	CU 10-22	1	2,750
	−10	AH-10	2	5,560	Built-up system[3]		12,620
15: Base	32	AH-9	4	6,880	CU 7½-12	2	4,100
	50	AH-3	3	2,530	CU 7½-12	2	3,350
	−10	AH-10	2	7,650	CU 10-22	6	15,200
Alternate	32	AH-9	4	6,880	CU 15-22	1	3,680
	50	AH-3	3	2,530	CU 10-22	1	2,600
	−10	AH-10	2	7,650	Built-up system[4]		13,820
16: Base	32	AH-9	3	4,520	CU 7½-12	2	6,300
	50	AH-3	3	2,260	CU 7½-12	2	4,304
Alternate	32	AH-9	3	4,520	CU 10-12	1	4,930
	50	AH-3	3	2,260	CU 10-12	1	3,590
17: Base	32	AH-9	2	4,140	CU 7½-12	2	4,750
	50	AH-3	2	1,770	CU 5-12	2	2,610
	−10	AH-10	1	3,120	CU 10-22	3	8,144
Alternate	32	AH-9	2	4,140	CU 15-12	1	3,480
	50	AH-3	2	1,770	CU 7½-12	1	1,990
	−10	AH-10	1	3,120	CU 7½-502	3	7,400
18: Base	32	AH-10	1	2,940	CU 5-12	2	2,480
	50	AH-3	1	1,055	CU 2-12	2	1,995
	−10	AH-10	1	2,980	CU 10-22	3	8,650
Alternate	32	AH-10	1	2,940	CU 7½-12	1	2,180
	50	AH-3	1	1,055	CU 5-12	1	1,680
	−10	AH-10	1	2,980	CU 7½-22	3	7,470
19: Base	40	AH-7	1	1,775	CU 5-12	2	2,630
	−10	AH-10	2	5,220	CU 10-22	6	15,350
Alternate	40	AH-7	1	1,775	CU 7½-12	1	1,990
	−10	AH-10	2	5,220	Built-up system[5]		14,050
20: Base	40	AH-6	2	2,402	CU 7½-12	2	3,980
Alternate	40	AH-6	2	2,402	CU 10-12	1	3,150
21: Base	72	AH-1	1	1,760	CU ½-12	2	1,140
	40	AH-6	1	1,106	CU 3-12	2	2,145
	−20	AH-18	1	1,210	CU 3-22	2	3,690
Alternate	72	AH-1	1	1,760	CU 1½-12	1	990
	40	AH-6	1	1,106	CU 7½-12	1	2,110
	−20	AH-18	1	1,210	CU 7½-22	1	2,765

See footnotes at end of table on page 66.

TABLE 23.—*Situation I, bill of materials and unit costs for refrigeration equipment*—Continued

Firm number and proposal	Room temperature	Air unit model number	Number required	Installed cost	Condensing unit model number	Number required	Installed cost[1]
	° F.		*Number*	*Dollars*		*Number*	*Dollars*
22: Base	72	AH-3	3	2,940	CU 5-12	2	2,648
	50	AH-5	2	1,802	CU 7½-12	2	4,086
	40	AH-6	2	2,372	CU 7½-12	2	4,086
	-10	AH-10	3	8,150	CU 10-502	6	15,600
Alternate	72	AH-3	3	2,940	CU 10-12	1	2,510
	50	AH-5	2	1,802	CU 10-12	1	2,520
	40	AH-6	2	2,372	CU 10-12	1	2,520
	-10	AH-10	3	8,150	Built-up system[6]		17,370
23: Base	72	AH-3	1	1,069	CU 2-12	2	2,256
	50	AH-6	1	1,486	CU 3-12	2	2,760
Alternate	72	AH-3	1	1,069	CU 3-12	1	1,380
	50	AH-6	1	1,486	CU 7½-12	1	2,040
24: Base	72	AH-1	1	624	CU ½-12	2	1,172
	50	AH-5	2	1,728	CU 7½-12	2	4,080
	-20	AH-9	1	2,020	CU 10-22	2	5,521
Alternate	72	AH-1	1	624	Incl. below		
	50	AH-5	2	1,728	CU 10-12	1	2,220
	-20	AH-9	1	2,020	CU 5-502	2	3,720
25: Base	-10	AH-10	2	5,560	CU 10-22	3	8,100
Alternate	-10	AH-10	2	5,560	CU 10-22	3	8,100
26: Base	50	AH-6	1	1,406	CU 5-12	2	3,150
	-10	AH-9	2	4,060	CU 7½-22	3	7,520
Alternate	50	AH-6	1	1,406	CU 7½-12	1	1,820
	-10	AH-9	2	4,060	CU 10-502	2	5,680
27: Base	40	AH-7	1	1,770	CU 5-12	2	2,885
	25	AH-9	2	3,880	CU 7½-12	2	5,250
Alternate	40	AH-7	1	1,770	CU 7½-12	1	1,765
	25	AH-9	2	3,880	CU 15-22	1	3,164

TABLE 23.—*Situation I, bill of materials and unit costs for refrigeration equipme*—Continued

Firm number and proposal	Room temperature	Air unit model number	Number required	Installed cost	Condensing unit model number	Number required	Insta cos
	° F.		*Number*	*Dollars*		*Number*	*Doll*
28: Base	72	AH-2	2	1,500	CU 3-12	2	2,4
	40	AH-7	2	3,390	CU 7½-12	2	3,5
Alternate	72	AH-2	2	1,500	CU 5-12	1	1,5
	40	AH-7	2	3,390	CU 15-12	1	2,5
30: Base	50	AH-7	1	1,511	CU 7½-12	2	3,5
	32	AH-10	2	5,300	CU 7½-12	2	4,1
	-10	AH-9	1	1,870	CU 5-22	3	6,6
Alternate	50	AH-7	1	1,511	CU 19-12	1	2,5
	32	AH-10	2	5,300	CU 15-12	1	2,6
	-10	AH-9	1	1,870	CU 10-22	2	6,31
31: Base	32	AH-10	2	4,980	CU 7½-12	2	4,12
Alternate	32	AH-10	2	4,980	CU 15-22	1	4,04
33: Base	32	AH-10	2	5,250	CU 7½-12	2	4,07
	-10	AH-9	1	1,955	CU 7½-22	2	4,12
Alternate	32	AH-10	2	5,250	CU 15-22	1	4,10
	-10	AH-9	1	1,955	CU 10-502	1	3,32

[1] Open type except as noted.

[2] Semihermetic.

[3] *R-502*—12.6 tons of refrigeration at -25° F. evaporator temperature. Two open-st reciprocating compressors with 25 hp. motors and starters. One air-cooled condenser with win pressure control. One refrigerant receiver with necessary valves and accessories.

[4] *R-502*—15.5 tons of refrigeration at -22° F. evaporator temperature. Two open-style recip cating compressors with 25 hp. motors and starters. One air-cooled condenser with wint pressure controls. One refrigerant receiver with necessary valves and accessories.

[5] *R-502*—14.5 tons of refrigeration at -22° F. evaporator temperature. Two open-sty reciprocating compressors with 25 hp. motors and starters. One air-cooled condenser with wint pressure control. One refrigerant receiver with necessary valves and accessories.

[6] *R-602*—18.1 tons of refrigeration at -21° F. evaporator temperature. One open-sty reciprocating compressor with 25 hp. motor and starter. One open-style reciprocating compress with 40 hp. motor and starter. One air-cooled condenser with winter pressure control. O refrigerant receiver with necessary valves and accessories.

TABLE 24.—*Situation I, Building No. 1, Summary of installed costs and owning and operating costs by firm*[1]

Base proposal:

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
1	2,916	1,103	70.00
2	2,596	502	52.80
3	10,005	7,983	249.95
4	14,540	9,997	312.45
5	17,402	8,325	340.80
6	13,807	7,917	420.85
7	27,831	12,303	517.00
8	15,530	8,843	307.77
9	22,350	13,887	618.43
10	15,690	8,944	334.29
		17,600 doors	
Totals	142,667	97,404	3,224.24

Amortization, insulation (20 yr. @ 6%)	97,404 × 0.08718 =	$ 8,492
Amortization, refrigeration (10 yr. @ 6%)	142,667 × 0.13587 =	10,384
Maintenance, insulation (2%/yr.)	97,404 × 0.02 =	1,948
Maintenance, refrigeration (10%/yr.)	142,667 × 0.10 =	14,268
Insurance, $1.81/thousand adjusted	240,071 × 0.00181 =	435
Taxes, $4.88/thousand adjusted	240,071 × 0.00488 =	1,172
Electric power cost		38,691
Annual owning and operating cost		$84,389

Alternate proposal:

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
1	2,416	1,103	70.00
2	2,026	502	52.80
3	9,215	7,983	249.95
4	14,540	9,997	312.35
5	15,740	8,325	340.80
6	13,807	7,917	420.85
7	25,400	12,303	517.00
8	13,550	8,843	307.77
9	22,350	13,887	618.43
10	13,700	8,944	334.29
		17,600 doors	
Totals	132,744	97,404	3,224.24

Amortization, insulation (20 yr. @ 6%)	97,404 × 0.08718 =	$ 8,492
Amortization, refrigeration (10 yr. @ 6%)	132,744 × 0.13587 =	18,036
Maintenance, insulation (2%/yr.)	97,404 × 0.02 =	1,948
Maintenance, refrigeration (10%/yr.)	132,744 × 0.10 =	13,274
Insurance, $1.81/thousand adjusted	230,148 × 0.00181 =	417
Taxes, $4.88/thousand adjusted	230,148 × 0.00488 =	1,123
Electric power cost		38,691
Annual owning and operating cost		$81,981

[1] Air conditioning not included.

TABLE 25.—*Situation I, Building No. 2, summary of installed costs and owning and operating costs by firm*[1]

Base proposal

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
11	10,302	5,689	180.25
12	9,500	10,735	196.55
13	26,631	15,822	535.25
14	36,206	20,263	699.36
15	39,710	20,847	763.58
16	17,384	14,013	342.95
17	24,534	10,305	458.46
18	20,002	6,704	377.52
		28,000 doors	
Totals	184,269	132,378	3,553.92

Amortization, insulation (20 yr. @ 6%)	132,378 × 0.08718 =	$11,541
Amortization, refrigeration (10 yr. @ 6%)	184,269 × 0.13587 =	25,037
Maintenance, insulation (2%/yr.)	132,378 × 0.02 =	2,648
Maintenance, refrigeration (10%/yr.)	184,269 × 0.10 =	18,427
Insurance, $1.81/thousand adjusted	316,647 × 0.00181 =	573
Taxes, $4.88/thousand adjusted	316,047 × 0.00488 =	1,545
Electric power cost		42,647
Annual owning and operating cost		$102,418

Alternate proposal

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
11	8,905	5,689	180.25
12	9,160	10,735	196.55
13	23,770	15,822	535.25
14	34,830	20,263	699.36
15	37,160	20,847	763.58
16	15,300	14,013	342.95
17	21,900	10,305	458.46
18	18,305	6,704	377.52
		28,000 doors	
Totals	169,330	132,378	3,553.92

Amortization, insulation (20 yr. @ 6%)	132,378 × 0.08718 =	$11,541
Amortization, refrigeration (10 yr. @ 6%)	169,330 × 0.13587 =	23,007
Maintenance, insulation (2%/yr.)	132,378 × 0.02	2,648
Maintenance, refrigeration (10%/yr.)	169,330 × 0.10	16,933
Insurance, $1.81/thousand adjusted	301,708 × 0.00181 =	546
Taxes, $4.88/thousand adjusted	301,708 × 0.00488 =	1,472
Electric power cost		42,647
Annual owning and operating cost		98,794

[1] Air conditioning not included.

TABLE 26.—*Situation I, Building No. 3, summary of installed costs and owning and operating costs by firm*[1]

Base proposal:

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
19	24,975	10,087	564.18
20	6,382	3,984	143.44
21	11,051	3,526	160.51
22	41,684	23,090	739.45
23	7,571	6,313	143.45
24	15,145	6,868	289.18
25	13,660	13,202	317.50
26	16,136	13,340	295.85
27	13,785	11,277	282.92
28	10,820	10,770	244.60
		27,395 doors	
Totals	161,209	129,852	3,181.08

Amortization, insulation (20 yr. @ 6%)	129,852 × 0.08718 =	$11,320
Amortization, refrigeration (10 yr. @ 6%)	161,209 × 0.13587 =	21,903
Maintenance, insulation (2%/yr.)	129,852 × 0.02 =	2,598
Maintenance, refrigeration (10%/yr.)	161,209 × 0.10 =	16,121
Insurance, $1.81/thousand adjusted	291,061 × 0.00181 =	527
Taxes, $4.88/thousand adjusted	291,061 × 0.00488 =	1,420
Electric power cost		38,173
Annual owning and operating cost		$92,062

Alternate proposal:

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
19	23,035	10,087	564.18
20	5,552	3,984	143.44
21	9,941	3,526	160.51
22	40,184	23,090	739.45
23	5,075	6,313	143.45
24	10,312	6,868	289.18
25	13,060	13,202	317.50
26	12,966	13,340	295.85
27	10,579	11,277	282.92
28	9,030	10,770	244.60
		27,395 doors	
Totals	141,234	129,852	3,181.08

Amortization, insulation (20 yr. @ 6%)	129,852 × 0.08718 =	$11,320
Amortization, refrigeration (10 yr. @ 6%)	141,234 × 0.13587 =	19,189
Maintenance, insulation (2%/yr.)	129,852 × 0.02 =	2,598
Maintenance, refrigeration (10%/yr.)	141,234 × 0.10 =	14,123
Insurance, $1.81/thousand adjusted	271,086 × 0.00181 =	491
Taxes, $4.88/thousand adjusted	271,086 × 0.00488 =	1,323
Electric power cost		38,173
Annual owning and operating cost		$87,217

[1] Air conditioning not included.

TABLE 27.—*Situation I, Building No. 4, summary of installed costs and owning and operating costs by firm*[1]

Base proposal:

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
29 (no refrigeration)			29.95
30	23,031	14,122	467.31
31	9,105	4,623	237.69
32 (no refrigeration)			103.20
33	15,395	11,653	350.67
34 (no refrigeration)			30.00
		5,890 (doors)	
Totals	47,531	36,288	1,218.82

Amortization, insulation (20 yr. @ 6%)	36,288 × 0.08718 =	$ 3,164
Amortization, refrigeration (10 yr. @ 6%)	47,531 × 0.13587 =	6,458
Maintenance, insulation (2%/yr.)	36,288 × 0.02 =	726
Maintenance, refrigeration (10%/yr.)	47,531 × 0.10 =	4,753
Insurance, $1.81/thousand adjusted	83,819 × 0.00181 =	152
Taxes, $4.88/thousand adjusted	83,819 × 0.00488 =	409
Electric power cost		14,626
Annual owning and operating cost		$30,288

Alternate proposal:

Firm number	Refrigeration cost	Insulation cost	Operating cost/month
	Dollars	*Dollars*	*Dollars*
29 (no refrigeration)			29.95
30	20,097	14,122	467.31
31	9,020	4,623	237.69
32 (no refrigeration)			103.20
33	14,625	11,653	350.67
34 (no refrigeration)			30.00
		5,890 (doors)	
Totals	43,742	36,288	1,218.82

Amortization, insulation (20 yr. @ 6%)	36,288 × 0.08718 =	$ 3,164
Amortization, refrigeration (10 yr. @ 6%)	43,742 × 0.13587 =	5,943
Maintenance, insulation (2%/yr.)	36,288 × 0.02 =	726
Maintenance, refrigeration (10%/yr.)	43,742 × 0.10 =	4,374
Insurance, $1.81/thousand adjusted	80,030 × 0.00181 =	145
Taxes, $4.88/thousand adjusted	80,030 × 0.00488 =	391
Electric power cost		14,626
Annual owning and operating cost		$29,369

[1] Air conditioning not included.

TABLE 28.—*Situation II, Building No. 1, insulation costs and electric power consumption caused by transmission-heat gains*

Insulation code[1] and room temperature	Q/A[2]	Area	Hours/year	Conversion (× 10^{-3})	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
° F.	*B.t.u./hr.-ft.²*	*Ft.²*	*Hours*	*Kw./B.t.u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
B	1.739	9,954	4,629	0.109	8,734	1.121	11,158
C	2.239	966	4,629	.109	1,091	1.121	1,083
D	2.248	88	5,886	.109	130	1.121	99
E	2.052	3,459	7,422	.109	5,742	1.244	4,303
J	2.19	4,410	8,760	.109	9,221	1.206	5,318
K	2.543	2,940	8,760	.109	7,138	1.206	3,546
L	2.544	88	8,760	.109	214	1.248	110
M	3.029	5,640	8,760	.109	16,306	1.329	7,496
M-1	1.729	4,596	8,760	.109	7,587	1.202	5,524
Ceiling:							
50	3.225	14,070	4,629	.109	22,900	1.196	16,828
45	3.146	1,610	4,629	.109	2,556	1.238	1,993
40	3.058	80	5,886	.109	157	1.238	99
32	3.037	5,805	7,422	.109	14,257	1.238	7,187
Floor:							
50	.832	14,070	8,760	.109	10,664	.211	2,969
45	1.255	1,610	8,760	.109	1,929	.253	407
40	1.508	80	8,760	.109	115	.296	24
32	1.918	5,805	8,760	.109	10,628	.409	2,374
Total					119,369		70,518

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 29.—*Situation II, Building No. 2, insulation costs and electric power consumption caused by transmission-heat gains*

Insulation code[1] and room temperature	Q/A[2]	Area	Hours/year	Conversion (× 10^{-3})	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
° F.	*B.t.u./hr.-ft.²*	*Ft.²*	*Hours*	*Kw./B.t.u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
B	1.739	5,369	4,629	0.109	4,711	1.121	6,019
E	2.052	8,333	7,422	.109	13,833	1.244	10,366
G	2.193	1,105	8,760	.214	4,543	1.527	1,687
M-1	1.729	4,581	8,760	.109	7,563	1.202	5,506
O-2	2.138	2,106	8,760	.214	8,441	1.527	3,216
O-4	1.763	1,105	8,760	.214	3,652	1.462	1,616
Ceiling:							
50	3.225	8,800	4,629	.109	14,322	1.196	10,525
32	3.037	24,120	7,422	.109	59,263	1.238	29,861
−10	2.516	2,080	8,760	.214	9,801	1.592	3,311
Floor							
50	.832	8,800	4,629	.109	3,694	.211	1,857
32	1.918	24,120	7,422	.109	37,412	.409	9,865
−10	2.603	2,080	8,760	.214	10,150	.664	1,381
Total					177,385		85,210

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 30.—*Situation II, Building No. 3, insulation costs and electric power consumption caused by transmission-heat gains*

Insulation code[1] and room temperature	Q/A[2]	Area	Hours/year	Conversion (× 10⁻³)	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
° F.	*B.t.u./hr.-ft.²*	*Ft.²*	*Hours*	*Kw./B.t.u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
A	1.417	4,200	1,776	0.109	1,152	0.833	3,499
B	1.739	2,420	4,629	.109	2,123	1.121	2,713
D	2.248	4,680	5,886	.109	6,750	1.121	5,246
F	2.248	960	7,422	.109	1,746	1.287	1,236
G	2.193	2,440	8,760	.214	10,031	1.527	3,726
I	1.303	1,700	8,760	.109	2,115	.876	1,489
J	2.19	3,040	8,760	.109	6,357	1.206	3,666
J-1	1.845	1,500	8,760	.109	2,642	1.163	1,745
L	2.544	1,900	8,760	.109	4,616	1.248	2,371
L-1	2.295	1,840	8,760	.109	4,032	1.206	2,219
N	3.076	700	8,760	.109	2,056	1.372	960
N-2	1.417	700	8,760	.109	947	1.202	841
N-3	1.575	560	8,760	.109	842	1.285	720
O	2.645	3,080	8,760	.214	15,273	1.625	5,005
O-1	2.561	440	8,760	.214	2,113	1.592	700
O-2	2.138	2,440	8,760	.214	9,782	1.527	3,726
O-3	1.924	1,800	8,760	.214	6,493	1.495	2,691
O-5	1.47	800	8,760	.214	2,205	1.462	1,170
P	2.73	160	8,760	.214	819	1.657	265
P-1	2.657	360	8,760	.214	1,793	1.625	585
P-2	2.281	240	8,760	.214	890	1.56	452
P-3	2.096	200	8,760	.214	786	1.527	305
Ceiling:							
72	2.79	6,124	1,776	.109	3,308	1.196	7,324
50	3.225	3,671	4,629	.109	5,974	1.196	4,391
40	3.058	6,513	5,886	.109	12,778	1.238	8,063
25	3.083	1,680	7,422	.109	4,191	1.362	2,288
−10	2.516	6,224	8,760	.214	29,356	1.592	9,909
−20	2.616	288	8,760	.214	1,412	1.625	468
Floor:							
50	.832	3,671	4,629	.109	1,541	.211	775
40	1.508	6,513	5,886	.109	6,301	.296	1,928
25	2.134	1,680	7,422	.109	2,900	.452	750
−10	2.603	6,224	8,760	.214	30,371	.664	4,133
−20	2.74	228	8,760	.214	1,479	.707	204
Total					184,174		85,572

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 31.—*Situation II, Building No. 4, insulation costs and electric power consumption caused by transmission-heat gains*

Insulation code[1] and room temperature	Q/A[2]	Area	Hours/year	Conversion ($\times 10^{-3}$)	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
° *F.*	*B.t.u./hr.-ft.*²	*Ft.*²	*Hours*	*Kw./B.t.u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
B	1.739	480	4,629	0.109	421	1.121	538
E	2.052	1,440	7,422	.109	2,390	1.244	1,791
G	2.193	960	8,760	.214	3,947	1.527	1,466
J	2.19	1,360	8,760	.109	2,844	1.206	1,640
M	3.029	2,940	8,760	.109	8,503	1.329	3,907
M-1	1.729	480	8,760	.109	792	1.202	577
O	2.645	2,180	8,760	.214	10,809	1.625	3,543
O-4	1.763	1,180	8,760	.214	3,900	1.462	1,725
Ceiling:							
50	3.225	816	4,629	.109	1,328	1.196	976
32	3.037	2,736	7,422	.109	6,722	1.238	3,387
−10	2.516	1,440	8,760	.214	6,792	1.592	2,292
Floor:							
50	.832	816	4,629	.109	343	.211	172
32	1.918	2,736	7,422	.109	4,245	.409	1,119
−10	2.603	1,440	8,760	.214	7,027	.664	956
Total					60,063		24,089

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 32.—*Situation II, Buildings Nos. 1, 2, 3, and 4, electric power consumption and costs*

Type load	Annual power consumed (kw.-hr./yr.)			
	Building No. 1	Building No. 2	Building No. 3	Building No. 4
Demand requirements:[1]				
Transmission loss	119,369	177,385	184,174	60,063
High-stage heat sources:				
Lights and people	36,400	163,000	48,000	7,000
Fan motors	181,000	76,800	45,000	17,900
Air changes and product	874,000	206,000	260,400	144,300
Low-stage heat sources:				
Lights and people		6,850	23,300	6,700
Fan motors		105,000	56,500	9,800
Air changes and product		520,000	521,600	64,000
Operating requirements:[2]				
High-stage:				
Lights	438,000	655,000	462,500	438,000
Fan motors	298,500	126,700	149,400	73,600
Low-stage:				
Lights (included above)				
Fan motors		88,200	81,900	12,300
Total annual kw.-hr. consumption	1,947,269	2,124,935	1,832,774	833,763
Total annual kw.-hr. consumption, four buildings				6,738,741
Average monthly kw.-hr. consumption, four buildings				561,600

Maximum demand:	*kw.*	*Power cost per month:*	
Lights	455.1	200 kw. @ 2.00/kw.	= $ 400
Fan motors	94.8	800 kw. @ 1.80/kw.	= 1,440
Compressors	774.5	400 kw. @ 1.65/kw.	= 660
Pumps	15.4	100,000 kw.-hr @ 0.01148	= 1,148
Condensers	60.3	461,600 kw.-hr. @ 0.006	= 2,770
		Average monthly charge	= $ 6,418
Total demand	1,400.0		

[1] (Hr./yr.) × conversion factor (kw./B.t.u./hr. × 10^{-3}) × Q (B.t.u./hr.) = Annual power consumed (kw.-hr./yr.)

[2] The hr./yr. are multiplied by watts to find the annual power consumption (kw.-hr./yr.). The "conversion factor" and "Q" are not required in this calculation.

TABLE 33.—*Situation II, equipment building, bill of materials and unit cost*

Quantity	Equipment	Instal cost
		Dollars
3	*Ammonia Compressors*, Nos 1, 2, and 3, complete with 200-hp., 720-r.p.m., 440/3/60-volt motors and part-winding starters, per specifications	24,410
1	*Ammonia Compressor*, No 4, complete with a 125-hp., 1170-r.p.m., 440/3/60-volt motor and part-winding starter, per specifications	14,290
1	*Ammonia Booster Compressor*, No. 5, complete with a 30-hp., 1170-r.p.m., 440/3/60-volt motor and part-winding starter, per specifications	7,300
1	*Ammonia Booster Compressor*, No. 6, complete with a 75-hp., 720-r.p.m., 440/3/60-volt motor and part-winding starter, per specifications	20,560
3	*Evaporative Condensers*, complete with two 10-hp fan motors and a 1½-hp. pump motor with starters, per specifications	14,508
[1]1	*Horizontal Shell-and-Tube Heat Exchanger*, 30" diameter × 16′ long, complete with automatic valves and a float switch level control, per specifications	9,430
1	*High-Pressure Ammonia Receiver*, 36" diameter × 16′ long, with saddles, valves, and gage glass, per specifications	2,510
1	*Low-Stage Pump Accumulator*, 36" diameter × 4′0" long on a 6′0" high × 12" diameter leg complete with oil still, per plans and specifications	3,680
1	*Intercooler, Gas and Liquid*, 48" diameter × 4′0" long on a 6′0" high × 24" diameter leg containing a liquid cooling coil, per plans and specifications	4,180
2	*Compressor Jacket Coolers*, R-11, per specifications	610
2	*Oil Receivers* 12" diameter × 8′ long, with liquid level indicators, per specifications	500
2	*Ammonia Pumps*, low stage, complete with valves, 1½ hp., motor and starter, per specifications	1,225
2	*Ammonia Pumps*, high stage, complete with valves, 5-hp. motor and starter, per specifications	1,275
1	*Continuous Automatic Purger*, complete with valves and fittings, per specifications	800
	Ammonia and Oil Costs	1,400
[2]2	*Water Pumps*, complete with valves, 10-hp. motor and starter, per specifications	1,350
	Total	108,035

[1] Includes proportionate share of interconnecting piping costs.

[2] These components are used for air conditioning and/or heating duty.

TABLE 34.—*Situation II, air-handling units, bill of materials and unit cost*[1]

Quantity	Equipment	Installed cost (each)
	BUILDING NO. 1	*Dollars*
1	*AH-2RX, Air-Handling Unit*, air defrost, for ammonia liquid recirculation, complete with pipe, valves, automatic controls, and thermostat, per plans and specifications	1,060
1	*AH-5RX, Air-Handling Unit*, equipped as above	1,065
2	*AH-6RX, Air-Handling Units*, equipped as above	1,390
16	*AH-7RX, Air-Handling Units*, equipped as above	2,150
4	*AH-13RX, Air-Handling Units*, hot-gas defrost, equipped as above	2,360
3	*AH-14RX, Air-Handling Units*, equipped as AH-13RX	3,940
	Summary Building No. 1:	
	Equipment, installed cost (less insulation)	$51,665 (includes $11,050 labor)
	Pipe insulation cost	$ 8,900 (includes $ 2,935 labor)
	Total cost	$60,565 (includes $13,985 labor)
	BUILDING NO. 2	
13	*AH-3RX, Air-Handling Units*, air defrost, for ammonia liquid recirculation, complete with pipe, valves, pipe insulation, automatic controls, and thermostat, per plans and specifications	1,200
2	*AH-11RX, Air-Handling Units*, hot-gas defrost, equipped as above	1,395
1	*AH-12RX, Air-Handling Unit*, equipped as AH-11RX	1,805
3	*AH-13RX, Air-Handling Units*, equipped as AH-11RX	2,670
2	*AH-14RX, Air-Handling Units*, equipped as AH-11RX	4,450
10	*AH-15RX, Air-Handling Units*, equipped as AH-11RX	2,170
4	*AH-16RX, Air-Handling Units*, equipped as AH-11RX	2,220
8	*AH-17RX, Air-Handling Units*, equipped as AH-11RX	2,350
	Summary Building No. 2:	
	Equipment, installed cost (less insulation)	$70,535 (includes $15,100 labor)
	Pipe insulation cost	$15,950 (includes $ 5,320 labor)
	Total cost	$86,485 (includes $20,420 labor)
	BUILDING NO. 3	*Dollars*
4	*AH-2RX, Air-Handling Units*, air defrost, for ammonia liquid recirculation, complete with pipe, pipe insulation, valves, automatic controls, and thermostat, per plans and specifications	1,255
3	*AH-3RX, Air-Handling Units*, equipped as above	1,670
4	*AH-6RX, Air-Handling Units*, equipped as above	2,180
7	*AH-7RX, Air-Handling Units*, equipped as above	3,360
3	*AH-11RX, Air-Handling Units*, hot-gas defrost, equipped as above	1,925
2	*AH-12RX, Air-Handling Units*, equipped as AH-11RX	2,500
5	*AH-13RX, Air-Handling Units*, equipped as AH-11RX	3,690
1	*AH-14RX, Air-Handling Unit*, equipped as AH-11RX	6,200
	Summary Building No. 3:	
	Equipment, installed cost (less insulation)	$60,375 (includes $17,954 labor)
	Pipe insulation cost	$17,200 (includes $ 5,740 labor)
	Total cost	$77,575 (includes $23,694 labor)
	BUILDING NO. 4	
1	*AH-7RX, Air-Handling Unit*, air defrost, for ammonia liquid recirculation, complete with pipe, pipe insulation, valves, automatic controls and thermostat, per plans and specifications	2,510
2	*AH-12RX, Air-Handling Units*, hot-gas defrost, equipped as above	2,230
3	*AH-14RX, Air-Handling Units*, equipped as AH-12RX	5,690
	Summary Building No. 4:	
	Equipment, installed cost (less insulation)	$15,070 (includes $ 4,150 labor)
	Pipe insulation cost	$ 8,970 (includes $ 3,015 labor)
	Total cost	$24,040 (includes $ 7,165 labor)

[1] Includes proportionate share of interconnecting piping costs.

TABLE 35.—*Situation III, Building No. 1, insulation costs and electric power consumption caused by transmission heat gains*

Insulation code[1] and room temperature	Q/A[2]	Area	Hours/year	Conversion ($\times 10^{-3}$)	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
° *F.*	*B.t.u./hr.-ft.*[2]	*Ft.*[2]	*Hours*	*Kw./B.t.u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
B	1.739	9,954	4,629	0.109	8,910	1.121	11,160
C	1.876	966	4,629	.109	932	1.163	1,125
D	1.883	88	5,886	.109	106	1.163	103
E	1.764	3,459	7,422	.109	4,930	1.287	4,450
J	1.927	4,410	8,760	.109	8,125	1.248	5,520
K	2.237	2,940	8,760	.109	6,375	1.248	3,670
L	2.271	88	8,760	.109	191	1.291	114
M	2.441	6,120	8,760	.109	14,300	1.414	8,660
M-1	1.453	4,116	8,760	.109	5,700	1.244	5,120
Ceiling:							
50	2.513	14,070	4,629	.109	17,850	1.281	18,000
45	2.799	1,610	4,629	.109	2,270	1.281	1,992
40	2.72	80	5,886	.109	164	1.281	99
32	2.70	5,805	7,422	.109	12,660	1.362	7,920
Floor:							
50	.832	14,070	8,760	.109	11,200	.211	2,960
45	1.01	1,610	8,760	.109	1,550	.296	477
40	1.261	80	8,760	.109	97	.338	27
32	1.648	5,805	8,760	.109	9,170	.452	2,622
Total					104,530		74,019

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 36.—*Situation III, Building No. 2, insulation costs and electric power consumption caused by transmission heat gains*

Insulation code[1] and room temperature	Q/A[2]	Area	Hours/year	Conversion ($\times 10^{-3}$)	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
° *F.*	*B.t.u./hr.-ft.*[2]	*Ft.*[2]	*Hours*	*Kw./B.t.u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
B	1.739	5,369	4,629	0.109	4,711	1.121	6,019
E	1.764	8,333	7,422	.109	11,892	1.287	10,725
G	1.753	1,105	8,760	.214	3,632	1.625	1,796
M-1	1.453	5,369	8,760	.109	7,680	1.244	6,679
O-2	1.834	2,106	8,760	.214	7,242	1.592	3,353
O-4	1.47	1,105	8,760	.214	3,046	1.527	1,687
Ceiling:							
50	2.513	8,800	4,629	.109	11,159	1.281	11,273
32	2.70	24,120	7,422	.109	52,685	1.362	32,851
−10	2.068	2,080	8,760	.214	8,054	1.69	3,515
Floor:							
50	.832	8,800	4,626	.109	3,694	.211	1,857
32	1.648	24,120	7,422	.109	32,158	.452	10,902
−10	2.232	2,080	8,760	.214	8,705	.749	1,558
Total					154,658		92,215

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

BLE 37.—*Situation III, Building No. 3, insulation costs and electric power consumption caused by transmission-heat gains*

Insulation code[1] and om temperature	Q/A[2]	Area	Hours/year	Conversion ($\times 10^{-3}$)	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
° *F.*	*B.t.u./hr.-ft.*[2]	*Ft.*[2]	*Hours*	*Kw./B t u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
alls:							
	0.976	4,200	1,776	0.109	793	0.876	3,679
	1.739	2,420	4,629	.109	2,075	1.121	2,713
)	1.883	4,680	5,886	.109	5,654	1.163	5,443
	1.971	960	7,422	.109	1,530	1.329	1,276
G	1.753	2,440	8,760	.214	8,019	1.625	3,965
	1.005	1,700	8,760	.109	1,632	.918	1,561
	1.927	3,040	8,760	.109	5,594	1.248	3,794
-1	1.593	1,500	8,760	.109	2,282	1.206	1,809
	2.271	1,900	8,760	.109	4,120	1 291	2,453
-1	2.019	1,840	8,760	.109	3,547	1 248	2,296
	2.777	700	8,760	.109	1,856	1.414	990
N-2	1.417	700	8,760	.109	947	1.202	841
N-3	2.46	560	8,760	.109	1,331	1.202	673
O	2.207	3,080	8,760	.214	12,744	1.722	5,304
O-1	2.112	440	8,760	.214	1,742	1.69	741
O-2	1.834	2,440	8,760	.214	8,389	1.592	3,884
O-3	1.629	1,800	8,760	.214	5,496	1.56	2,808
O-5	1.225	800	8,760	.214	1,837	1.527	1,222
P	2.301	160	8,760	.214	690	1.755	281
P-1	2.216	360	8,760	.214	1,496	1.722	620
P-2	1.978	240	8,760	.214	890	1.625	390
P-3	1.798	200	8,760	214	674	1.592	318
eiling:							
72	2.79	6,124	1,776	.109	3,308	1.196	7,324
50	2.513	3,671	4,629	.109	4,654	1.281	4,703
40	2.72	6,513	5,886	.109	11,369	1.281	8,343
25	2.775	1,680	7,422	.109	3,771	1.404	2,359
10	2.068	6,224	8,760	.214	24,118	1.69	10,519
-20	2.176	288	8,760	.214	1,175	1.722	496
oor:							
50	.832	3,671	4,629	.109	1,541	.211	775
40	1.261	6,513	5,886	.109	5,269	.338	2,201
25	1.87	1,680	7,422	.109	2,542	.494	830
10	2.232	6,224	8,760	.214	26,044	.749	4,662
20	2.375	288	8,760	.214	1,282	.792	228
Total					158,411		89,501

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 38.—*Situation III, Building No. 4, insulation costs and electric power consumption caused by transmission-heat gains*

Insulation code[1] and room temperature	Q/A[2]	Area	Hours/year	Conversion (× 10⁻³)	Power consumed	Insulation costs: Cost per square foot	Insulation costs: Installed cost
°F	*B.t.u./hr.-ft.²*	*Ft.²*	*Hours*	*Kw./B.t.u./hr.*	*Kw.-hr./yr.*	*Dollars*	*Dollars*
Walls:							
B	1.739	480	4,629	0.109	421	1.121	538
E	1.764	1,440	7,422	.109	2,055	1.287	1,853
G	1.753	960	8,760	.214	3,155	1.625	1,560
J	1.927	1,360	8,760	.109	2,503	1.248	1,697
M	2.441	2,940	8,760	.109	6,853	1.414	4,157
M-1	1.453	480	8,760	.109	666	1.244	597
O	2.207	2,180	8,760	.214	9,019	1.722	3,754
O-4	1.47	1,180	8,760	.214	3,268	1.527	1,802
Ceiling:							
50	2.513	816	4,629	.109	1,035	1.281	1,045
32	2.70	2,736	7,422	.109	6,031	1.362	3,726
−10	2.068	1,440	8,760	.214	5,583	1.69	2,434
Floor:							
50	.832	816	4,629	.211	349	.211	172
32	1.648	2,736	7,422	.452	3,647	.452	1,237
−10	2.232	1,440	8,760	.749	6,024	.749	1,079
Total					50,609		25,651

[1] Insulation code refers to different thicknesses and types of insulation and temperature differences across wall, as shown in tables 3, 5, and 7.
[2] WTD Q/A is used for outside walls and roofs or ceilings. Standard Q/A is used for interior walls and floors.

TABLE 39.—*Situation III, Building No. 1, electric power consumption and costs*

Type load	Annual power consumed (kw.-hr./yr.)
Demand requirements:[1]	
Transmission loss	104,500
High-stage heat sources:	
Lights and people	36,300
Fan motors[2]	181,100
Air changes and product	871,800
Operating requirements:[3]	
Lights	438,000
Fan motors[4]	326,700
Total annual kw.-hr. consumption	1,958,400
Average monthly kw.-hr. consumption	163,200

Maximum demand:	kw.
Lights	100.0
Fan motors	37.3
Compressors	252.0
Pumps	5.6
Evaporator condensers	41.0
Total demand	435.9

Power cost/month:	
Demand:	
200 kw. @ $2.00/kw.	= $ 400
235.9 kw. @ $1.80/kw.	= 425
Energy:	
163,200 kw.-hr.	= 1,622
Average monthly cost	= 2,447
Yearly power cost	= $29,364

[1] (Hr./yr.) × conversion factor (kw./B.t.u./hr. × 10⁻³) × Q (B.t.u./hr.) = Annual power consumed (kw.-hr./yr.).
[2] Fan motor hp. in space was estimated.
[3] The hr./yr. are multiplied by watts to find the annual power consumption (kw.-hr./yr.) The "conversion factor" and "Q" are not required in this calculation.
[4] Based on actual selections of air-handling units.

TABLE 40.—*Situation III, Building No. 2, electric power consumption and costs*

Type load	Annual power consumed (kw.-hr./yr.)
Demand requirements:[1]	
Transmission loss	154,700
High-stage heat sources:	
Lights and people	163,600
Fan motors[2]	76,800
Air changes and product	205,300
Low-stage heat sources:	
Lights and people	6,900
Fan motors[2]	104,900
Air changes and product	521,700
Operating requirements:[3]	
Lights	655,700
Fan motors[4]	237,400
Total annual kw.-hr. consumption	2,127,000
Average monthly kw.-hr. consumption	177,300

Maximum demand:	*kw.*	*Power cost/month:*	
Lights	149.7	Demand:	
Fan motors	27.1	200 kw. @ $2.00/kw.	= $ 400
Compressors	256.5	276.2 kw. @ $1.80/kw.	= 497
Pumps	7.0	Energy:	
Evaporator condensers	35.9	177,300 kw.-hr.	= 1,728
Total demand	476.2	Average monthly cost	= 2,625
		Yearly power cost	= $31,500

[1] (Hr./yr.) × conversion factor (kw./B.t.u./hr. × 10^{-3}) × Q (B.t.u./hr.) = Annual power consumed (kw.-hr./yr.).

[2] Fan motor hp. in space was estimated.

[3] The hr./yr. are multiplied by watts to find the annual power consumption (hw.-hr./yr.). The "conversion factor" and "Q" are not required in this calculation.

[4] Based on actual selections of air-handling units.

TABLE 41.—*Situation III, Building No. 3, electric power consumption and costs*

Type load	Annual power consumed (kw.-hr./yr.)
Demand requirements:[1]	
Transmission loss	158,400
High-stage heat sources:	
Lights and people	48,100
Fan motors[2]	45,000
Air changes and product	232,900
Low-stage heat sources:	
Lights and people	23,200
Fan motors[2]	56,400
Air changes and product	549,500
Operating requirements:[3]	
Lights	497,500
Fan motors[4]	177,800
Total annual kw.-hr. consumption	1,788,800
Average monthly kw.-hr. consumption	149,100

Maximum demand:	*kw.*	*Power cost/month:*	
Lights	109.5	Demand:	
Fan motors	25.4	200 kw. @ $2.00/kw.	= $ 400
Compressors	257.0	228.7 kw. @ $1.80/kw.	= 412
Pumps	7.0	Energy:	
Evaporator condensers	29.8	149,100 kw.-hr.	= 1,517
Total demand	428.7	Average monthly cost	= 2,329
		Yearly power cost	= $27,948

[1] (Hr./yr.) × conversion factor (kw./B.t.u./hr. × 10^{-3}) × Q (B.t.u./hr.) = Annual power consumed (kw.-hr./yr.).

[2] Fan motor hp. in space was estimated.

[3] The hr./yr. are multiplied by watts to find the annual power consumption (kw.-hr./yr.) The "conversion factor" and "Q" are not required in this calculation.

[4] Based on actual selections of air-handling units.

TABLE 42.—*Situation III, Building No. 4, electric power consumption and costs*

Type load	Annual power consumed (kw.-hr./yr.)
Demand requirements.[1]	
Transmission loss	50,600
High-stage heat sources:	
Lights and people	7,300
Fan motors[2]	17,900
Air changes and product	144,400
Low-stage heat sources:	
Lights and people	6,700
Fan motors[2]	9,700
Air changes and product	64,000
Operating requirements:[3]	
Lights	438,000
Fan motors[4]	73,500
Total annual kw.-hr. consumption	812,100
Average monthly kw.-hr. consumption	67,700

Maximum demand:	*kw.*	*Power cost/month:*		
Lights	100.0	Demand:		
Fan motors	10.5	200 kw. @ $2.00/kw.	=	$ 400
Compressors	102.5	26.1 kw. @ $1.80/kw.	=	47
Pumps	2.8	Energy:		
Evaporator condensers	10.3	67,700 kw.-hr.	=	825
Total demand	226.1	Average monthly cost	=	1,272
		Yearly power cost	=	$15,264

[1] (Hr./yr.) × conversion factor (kw./B.t.u./hr. × 10^{-3}) × Q (B.t.u./hr.) = Annual power consumed (kw.-hr./yr.).

[2] Fan motor hp. in space was estimated.

[3] The hr./yr. are multiplied by watts to find the annual power consumption (kw.-hr./yr.) The "conversion factor" and "Q" are not required in this calculation.

[4] Based on actual selections of air-handling units.

TABLE 43.—*Situation III, Building No. 1, bill of materials and unit cost*[1]

Quantity	Equipment	Installed cost (each)
		Dollars
2	*Ammonia Compressors*, Nos 1 and 3, complete with 60-hp., 1170-r p.m., 440/3/60-volt motors and part-winding starters, per specifications	6,200
2	*Ammonia Compressors*, Nos 2 and 4, complete with 75-hp., 1170-r p m., 440/3/60-volt motors and part-winding starters, per specifications	7,320
2	*Evaporative Condensers*, Nos. 1 and 2, complete with 15-hp. fan motors and starters, 1½-hp. pumps and starters, per specifications	6,670
1	*Evaporative Condenser*, No. 3, complete with a 10-hp. fan motor and starter, 1-hp. pump and starter, per specifications	5,000
1	*High-Pressure Ammonia Receiver*, 30″ diameter × 16′ long, with stands, gage glass, and valves, per specifications	2,240
[2]1	*Horizontal Shell-and-Tube Water Chiller* (Heat Exchanger), 14″ diameter × 16′ long, and automatic control valves, per specifications	1,900
1	*Pump Accumulator*, 36″ diameter × 4′0″ long on a 12″ diameter × 6′0″ high leg, per plans and specifications	2,200
2	*Liquid Ammonia Pumps*, complete with a 3-hp motor, starter, and valves, per specifications	625
2	*Compressor Jacket Coolers*, R-11, complete with valves and fittings, per specifications	435
	Ammonia and Oil Costs	320
2	*Oil Receivers*, Nos. 1 and 2, 10″ diameter × 4′ long, per specifications	360
[2]2	*Water Pumps*, complete with 3-hp. motor and starter, per specifications	400
1	*Continuous Automatic Purger*, per specifications	700
1	*AH-2RX, Air-Handling Unit*, air defrost, for ammonia liquid recirculation, complete with pipe, valves, pipe insulation, automatic controls, and thermostat, per plans and specifications	1,060
1	*AH-5RX, Air-Handling Unit*, equipped as AH-2RX	1,065
2	*AH-6RX, Air-Handling Units*, equipped as AH-2RX	1,300
16	*AH-7RX, Air-Handling Units*, equipped as AH-2RX	2,150
4	*AH-13RX, Air-Handling Units*, hot-gas defrost, equipped as above	2,360
3	*AH-14RX, Air-Handling Units*, equipped as AH-13RX	3,940

[1] Includes proportionate share of interconnecting piping costs.
[2] These components are used for air conditioning, heating, or both.

TABLE 44.—*Situation III, Building No. 2, bill of materials and unit cost*[1]

Quantity	Equipment	Installed cost (each)
		Dollars
1	*Ammonia Compressor*, No. 1, high-stage, complete with 100-hp., 1170-r.p.m., 440/3/60-volt motor and part-winding starter, per specifications	7,600
1	*Ammonia Compressor*, No. 2, high-stage, complete with 125-hp., 1170-r.p.m., 440/3/60-volt motor and part-winding starter, per specifications	14,000
1	*Ammonia Compressor*, No. 3, high- or low-stage, complete with 60-hp, 1170-r.p.m, 440/3/60-volt motor and part-winding starter, per specifications	6,780
2	*Ammonia Booster-Compressors*, Nos. 4 and 5, low-stage, complete with 25-hp., 1170-r p.m., 440/3/60-volt motor and part-winding starter, per specifications	7,500
2	*Evaporative Condensers*, Nos. 1 and 2, complete with 10-hp. fan motor and starter, 1-hp. pump and starter, per specifications	5,000
1	*Evaporative Condenser*, No. 3, complete with 15-hp. motor and starter, 1½-hp. pump and starter, per specifications	6,650
1	*High-Pressure Ammonia Receiver*, 36" diameter × 16' long, with stands, gage glass, and valves, per specifications	2,340
[2]1	*Horizontal Shell-and-Tube Water Chiller* (Heat Exchanger), 14" diameter × 16' long, and automatic control valves, per specifications	1,990
1	*Pump Accumulator*, low-stage, 30" diameter × 4'0" long on a 12" diameter × 6'0" high leg, per plans and specifications	1,850
2	*Liquid Ammonia Pumps*, high-stage, complete with 3-hp. motor with starter and valves, per specifications	925
2	*Liquid Ammonia Pumps*, low-stage, complete with 1½-hp. motor with starter and valves, per specifications	925
	Ammonia and Oil Costs	670
[2]2	*Water Pumps*, complete with 3-hp. motor and starter, per specifications	400
1	*Continuous Automatic Purger*, complete with valves and fittings, per specifications	700
1	*Intercooler*, gas-and-liquid, 36" diameter × 4'0" long on a 24" diameter × 6'0" high leg containing a liquid cooling coil, per plans and specifications	2,550
2	*Compressor Jacket Coolers*, R-11, complete with valves and fittings, per specifications	435
2	*Oil Receivers*, 10" diameter × 4' long, complete per specifications	360
13	*AH-8RX, Air-Handling Units*, air defrost, for ammonia liquid recirculation, complete with pipe, valves, pipe insulation, automatic controls, and thermostat, per plans and specifications	1,200
2	*AH-11RX, Air-Handling Units*, hot-gas defrost, for ammonia liquid recirculation, complete with pipe, valves, insulation, automatic controls, and thermostat, per plans and specifications	1,395
1	*AH-12RX, Air-Handling Unit*, equipped as AH-11RX	1,805
3	*AH-13RX, Air-Handling Units*, equipped as AH-11RX	2,670
2	*AH-14RX, Air-Handling Units*, equipped as AH-11RX	4,450
10	*AH-15RX, Air-Handling Units*, equipped as AH-11RX	2,170
4	*AH-16RX, Air-Handling Units*, equipped as AH-11RX	2,220
8	*AH-17RX, Air-Handling Units*, equipped as AH-11RX	2,350

[1] Includes proportionate share of interconnecting piping costs.
[2] These components are used for air conditioning, heating, or both.

TABLE 45.—*Situation III, Building No. 3, bill of materials and unit cost*[1]

Quantity	Equipment	Installed cost (each)
		Dollars
3	*Ammonia Compressors*, Nos. 1, 2, and 3, high-stage, complete with 75-hp., 1170-r.p.m., 440/3/60-volt motors and part-winding starters, per specifications	8,750
1	*Ammonia Compressor*, No. 4, high- or low-stage, complete with 50-hp., 1170-r.p.m., 440/3/60-volt motor and part-winding starter, per specifications	6,550
2	*Ammonia Booster Compressors*, Nos. 5 and 6, low-stage, 25-hp., 1170-r.p.m., 440/3/60-volt motors and part-winding starters, per specifications	7,600
2	*Evaporative Condensers*, No. 1 and 2, complete with 15-hp. fan motors and starters, 1-hp. pump and starter, per specifications	6,650
1	*High-Pressure Ammonia Receiver*, 30″ diameter × 16′ long, complete with stands, gage glass, and valves, per specifications	1,890
[2]1	*Horizontal Shell-and-Tube Water Chiller* (Heat Exchanger), 14″ diameter × 16′ long, complete with level and automatic control valves per specifications	1,990
1	*Pump Accumulator*, low-stage, 30″ diameter × 4′ long on a 12″ diameter × 5′ high leg, per plans and specifications	2,120
1	*Intercooler*, gas-and-liquid, 30″ diameter × 4′ long on an 18″ diameter × 5′ high leg containing a liquid cooling coil, per plans and specifications	2,460
1	*Continuous Automatic Purger*, complete with valves and fittings, per specifications	600
2	*Liquid Ammonia Pumps*, high-stage, complete with 3-hp. motor and starter, valves, per specifications	925
2	*Liquid Ammonia Pumps*, low-stage, complete with 1½-hp. motor and starter, valves, per specifications	925
2	*Compressor Jacket Coolers*, R-11, complete with valves and fittings, per specifications	610
2	*Oil Receivers*, 10″ diameter × 4′ long, per specifications	360
[2]2	*Water Pumps*, complete with 3-hp. motor and starter, per specifications	400
	Ammonia and oil costs	750
4	*AH-2RX, Air-Handling Units*, air defrost, for liquid ammonia recirculation, complete with pipe, pipe insulation, valves, automatic controls, and thermostat, per plans and specifications	1,225
3	*AH-3RX, Air-Handling Units*, equipped as above	1,670
4	*AH-6RX, Air-Handling Units*, equipped as above	3,360
7	*AH-7RX, Air-Handling Units*, equipped as above	2,180
3	*AH-11RX, Air-Handling Units*, hot-gas defrost, equipped as above	1,925
2	*AH-12RX, Air-Handling Units*, hot-gas defrost, equipped as above	2,500
5	*AH-13RX, Air-Handling Units*, hot-gas defrost, equipped as above	3,690
1	*AH-14RX, Air-Handling Unit*, hot-gas defrost, equipped as above	6,200

[1] Includes proportionate share of interconnecting piping costs.
[2] These components are used for air conditioning, heating, or both.

TABLE 46.—*Situation III, Building No. 4, bill of materials and unit cost*[1]

Quantity	Equipment	Installed cost (each)
		Dollars
2	*Ammonia Compressors*, Nos. 1 and 2, high-stage, complete with 50-hp., 1015-r.p.m., 440/3/60-volt motors and part-winding starters, per specifications	6,700
1	*Ammonia Compressor*, No. 3, high-stage or low-stage, complete with 40-hp., 875-r.p.m., 440/3/60-volt motor and part-winding starter, per specifications	5,300
1	*Ammonia Booster Compressor*, No. 4, low-stage, complete with 10-hp., 725-r.p.m., 440/3/60-volt motor and voltage starter, per specifications	4,500
1	*Evaporative Condenser*, complete with 10-hp. fan motor and starter, 1-hp. pump motor and starter, per specifications	6,800
1	*High-Pressure Ammonia Receiver*, 24" diameter × 12' long complete with stands, gage glass, and valves, per specifications	980
1	*Horizontal Shell-and-Tube Water Chiller* (Heat Exchanger), 14" diameter × 16' long, complete with level and automatic control valves, per specifications	1,800
1	*Suction Trap*, low-stage, 12" diameter × 5' long with an internal liquid coil, per plans and specifications	620
1	*Intercooler*, gas-and-liquid, 12" diameter × 7' long containing a liquid cooling coil, per plans and specifications	640
2	*Compressor Jacket Coolers*, R-11, complete with valves and fittings, per specifications	335
2	*Oil Receivers*, 10" diameter × 4' long, complete, per specifications	260
1	*Continuous Automatic Purger*, complete with valves and fittings, per specifications	500
	Ammonia and Oil Costs	375
[2]2	*Water Pumps*, complete with a 3-hp. motor and starter, per specifications	400
1	*AH-7DX, Air-Handling Unit*, air defrost, for direct expansion system, complete with pipe, pipe insulation, valves, automatic controls, and thermostat, per plans and specifications	2,410
1	*AH-12DX, Air-Handling Unit*, hot-gas defrost, equipped as AH-7DX above	2,200
1	*AH-13DX, Air-Handling Unit*, equipped as AH-12RX	3,480
3	*AH-14DX, Air-Handling Units*, equipped as AH-12RX	5,690

[1] Includes proportionate share of interconnecting piping costs.
[2] These components are used for air conditioning, heating, or both.

TRANSLATION OF METHOD TO OTHER AREAS

This report has compared different combinations of insulation and refrigeration equipment for a hypothetical food distribution center in Chicago, Ill. The figures can be adapted to another area by considering the differences in refrigeration and insulation needed and in the costs of services and materials. By the same methods of calculation, the total cost can be figured for installing, owning, and operating these same refrigeration systems in any desired city or area. This section discusses the factors that would cause cost differentials between locations.

The major differences between areas are in (1) labor costs, (2) energy costs, and (3) climate.

Labor Costs

Differences in construction labor costs can be determined by multiplying the total installation labor cost by the ratio of the cost of local construction labor to the cost of construction labor in Chicago as used in this study ($8.05). The total installation labor cost for one central system (Situation II) in Chicago is $233,575 (table 13). The total cost of labor should include consideration of overhead, fringe benefits, and profit.

Electric Energy Costs

The base Chicago annual energy cost ($77,016 for Situation II, table 13) is first corrected for any difference in refrigeration load due to climate effect. This new total is then corrected for a difference in energy cost by multiplying it by the ratio of the "local energy cost" to the Chicago energy cost (0.01129 $/kw.).

The "local energy cost" is determined as an average value ($/kw.) based on the average monthly consumption for the type of refrigeration system considered.

Climate Effects

Differences in climate change the amount of insulation required in the ceiling and outside walls, as well as the size of the refrigeration system and the total refrigeration needed to compensate for the heat gains through these surfaces. The internal heat loads (from products, air changes, and miscellaneous loads) are assumed to remain the same from city to city. The only item that affects the size of the refrigeration system is the amount of heat gained through the wall, floor, and ceiling surfaces. The total effect of installing a different thickness of insulation and a different size of refrigeration system is relatively small compared with the overall costs, so quick hand calculations are sufficiently accurate.

This section contains sufficient information for hand calculating the change in insulation, equipment, and operating costs caused by differences in climate. The only decision necessary is whether to use a packaged unit refrigeration system, one central system, or four central systems. Specific information on these costs has been included for the cities of Atlanta, Boston, Chicago, Houston, Los Angeles, Orlando, Philadelphia, Phoenix, Seattle, and St. Louis.

There are five steps in securing sufficient data to which the cost information developed for Chicago can be applied. The five steps are:

1. Determine the type of refrigeration system to be used. This can be packaged units (Situation I), one central (Situation II), or four centrals (Situation III).

2. Determine the hours of operation per year during which the outside temperature exceeds the storage room design temperature. The weather data, including hours of sunshine, is available from the local weather bureau. It is tabulated and plotted on a curve similar to the one in figure 10, which represents temperatures used for wall surfaces only. A similar curve can be plotted for the roof, but the tabulation must include a penalty for the sunshine hours and load. See table 47 for a sample worksheet as tabulated for Atlanta, Ga.

Information in the upper part of table 47 is furnished by the Weather Bureau. It shows the total hours of sunshine and of outside atmospheric temperatures for each month of the year. The bottom part of the table shows the hours of various roof temperatures, for each month of the year, that were calculated from the Weather Bureau information in the upper part.

The purpose in calculating the hours in the bottom part of the table was to make adjustments for the increased heat on the roof caused by sunshine. This increase is significant and often pronounced. The effect of sunshine on a sidewalk is a good illustration. On a cloudy day, when the outside atmospheric temperature is about 80° F., the sidewalk temperature feels about the same as the air around it. At the same atmospheric temperature, but with the sun shining, the sidewalk is much hotter. Roof temperature reacts in the same way, and adjustments must be made to include this increased heat.

The hours shown in the upper part of the table did not necessarily occur in sequence or in daily increments. They are totals for the month as recorded by the Weather Bureau. Whether or not they were concurrent is unimportant, it is the cumulative buildup of differences between inside building temperature and outside building temperature that matters.

Our problem is to set apart and make temperature adjustments for the hours that the sun was shining. This was done through the use of two sets of temperature ranges that appear in the left column of the bottom part of the table. One set includes a 45° "penalty" factor, the other does not. The set bearing the penalty is used to record the hours of the month that the sun was shining. The set without the penalty is used to record the hours of the month that the sun did not shine.

To simplify the determination as to when the sun was shining and when not shining, it was assumed that the sun was shining only during the upper temperature ranges. This did not always happen, of course, but for our purposes this assumption is satisfactory.

The mechanics of the adjustment calculations are simple. Always consider that the sun was shining during the upper temperature ranges only. Work month by month, first subtracting from the total hours of sunshine the number of hours at which the outside atmospheric temperature range was highest. Second, subtract from the remaining hours of sunshine the number of hours in the next lower outside atmospheric temperature range, etc., until all the hours of sunshine are used up. All hours of sunshine are penalized. All hours without sunshine are not penalized.

In September, for example, there was a total of 375 hours of sunshine. There were 126 hours that the outside temperature reached 80° F. to 90°, the highest temperatures reached that month. Assuming that the sun was shining during the upper temperature ranges, we must penalize these 126 hours to adjust for the added heat from the sun. Therefore, they are entered in the bottom part of the table opposite 125° to 135°, which is the 80° to 90° range with the 45° penalty added. This accounts for 126 of the 375 hours of sunshine, and leaves a balance of 249 (375 − 126 = 249) hours of sunshine.

To account for all hours of sunshine in September, we now look to the next lower outside temperature range (70° to 80°). The weather bureau recorded 294 hours at 70° F. to 80°. Applying our assumption, the sun was shining for 249 of these 294 hours. These 249 hours of sunshine, therefore, are entered in the bottom part of the table opposite 115° to 125° (70° to 80° + 45°).

We now have a balance of 45 hours (294 − 249 = 45). These are hours that the sun did not shine, so they should not be penalized. These hours, then, are entered in the bottom part of the table opposite 70° to 80° (without penalty).

The other hours (276, 21, and 3), which are at lower outside temperature ranges, are merely transposed without penalty to the bottom part of the table since they, too, occurred when the sun was not shining.

This approach is not 100-percent accurate; however, the results are accurate enough for our purposes.

3. Determine the weighted-temperature-difference (WTD) that exists across each outside wall, roof, or ceiling surface. The method of computing this temperature difference has been explained in detail in "The Weighted Temperature–Hour Approach Developed for This Study."

TABLE 47.—*Sample worksheet—roof-temperature data, Atlanta, Ga.*

Item	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.[3]	Oct.	Nov.	Dec.	Total
Data obtained from Weather Bureau:	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*	*Hours*
Total hours of sunshine	250	230	260	310	375	300	325	330	*375*	340	300	250	
Outside (atmospheric) temperature (° F.).													
90° and above								12					12
80° to 90°				30	144	96	189	249	*126*		9		843
70° to 80°		6	12	159	285	279	423	375	*294*	42	114		1,989
60° to 70°	48	87	129	288	264	330	132	108	276	162	252	39	2,115
50° to 60°	201	120	213	150	51	15			21	285	246	186	1,488
40° to 50°	207	189	183	90					3	168	99	279	1,218
30° to 40°	186	162	171	3						54	24	204	804
20° to 30°	75	102	36							9		36	258
Below 20°	27	6											33
Calculated data:													
Outside wall temperature (° F.)[1]													
Roof temperature (° F.):													
With penalty:[2]													
135° and above (90° + 45°)								12					12
125° to 135° (80° to 90° + 45°)				30	144	96	189	249	*126*		9		843
115° to 125° (70° to 80° + 45°)		6	12	159	231	204	136	69	*249*	42	114		1,222
105° to 115° (60° to 70° + 45°)	48	87	129	121						162	177	39	763
95° to 105° (50° to 60° + 45°)	201	120	110							136		186	762
85° to 95° (40° to 50° + 45°)	1	17										25	43
75° to 85° (30° to 40° + 45°)													
65° to 75° (20° to 30° + 45°)													
Below 65° (20° + 45°)													
Without penalty:													
90° and above													
80° to 90°													
70° to 80°					54	75	287	306	*45*				767
60° to 70°				167	264	330	132	108	276		75		1,352
50° to 60°			94	150	51	15			21	149	246		726
40° to 50°	206	172	183	90					3	168	99	254	1,175
30° to 40°	186	162	171	3						54	24	204	804
20° to 30°	75	102	36							9		36	258
Below 20°	27	6											33

[1] Outside-wall temperature data assumed to be the same as that for outside atmospheric temperature.
[2] 45° extra temperature added to adjust for heat from the sun.
[3] Italicized numbers are explained on page 85.

4. Select the optimum thickness of insulation. For the internal walls and floors, this insulation thickness will be the same as used in Chicago, because the load exists for the entire year (8,760 hours) and the weighted-temperature-difference between each pair of rooms is the same.

The optimum-insulation computer program was operated specifically to find the temperature range over which each insulation thickness would be an optimum for a specific number of hours of operation per year. All cost factors and thermal resistances were the same as those used for Chicago. The results of these computer runs are illustrated in figures 43 through 51. For each type of refrigeration system, the optimum insulation thickness is shown for outside walls and ceilings of storage rooms above 32° F., using expanded polystyrene insulation, and for outside walls and ceilings of storage rooms 32° and below, using fibrous glass insulation.

The graphs for expanded-polystyrene insulation are based on 45° F. room calculations, and the fibrous-glass insulation graphs on −10° room calculations. The results shown on the graphs are thicknesses within one-half inch of the optimum for rooms above 32°, and within 1 inch of the optimum for the lower temperature rooms. Because the optimum-cost curve is relatively flat at the minimum (fig. 8), the difference is of little consequence. Minor variations in the refrigeration equipment cost or operating cost used as input to the computer have little effect upon the final thicknesses. Even major variations changed the final thicknesses less than 1 inch.

5. Determine the heat gain (Q) through each surface in B.t.u./ft.^{2}–hr. These values were obtained from the computer runs used in the preceding step and are illustrated in figures 52 and 53.

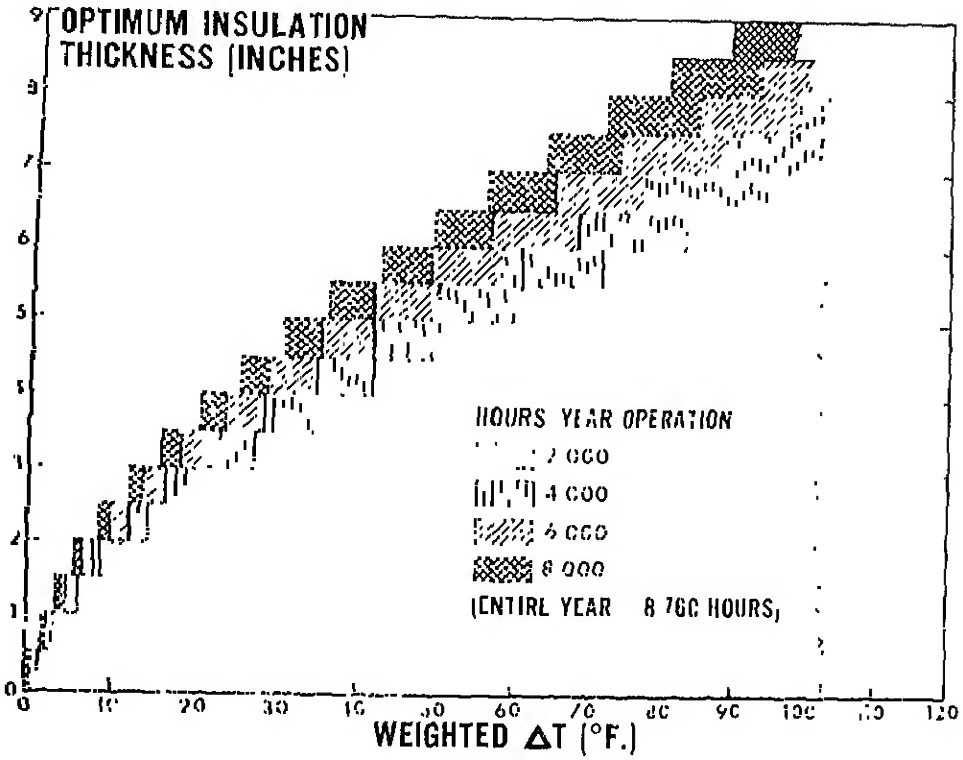


FIGURE 43.—Optimum thickness of expanded-polystyrene insulation for outside walls of rooms above 32° F.—package systems.

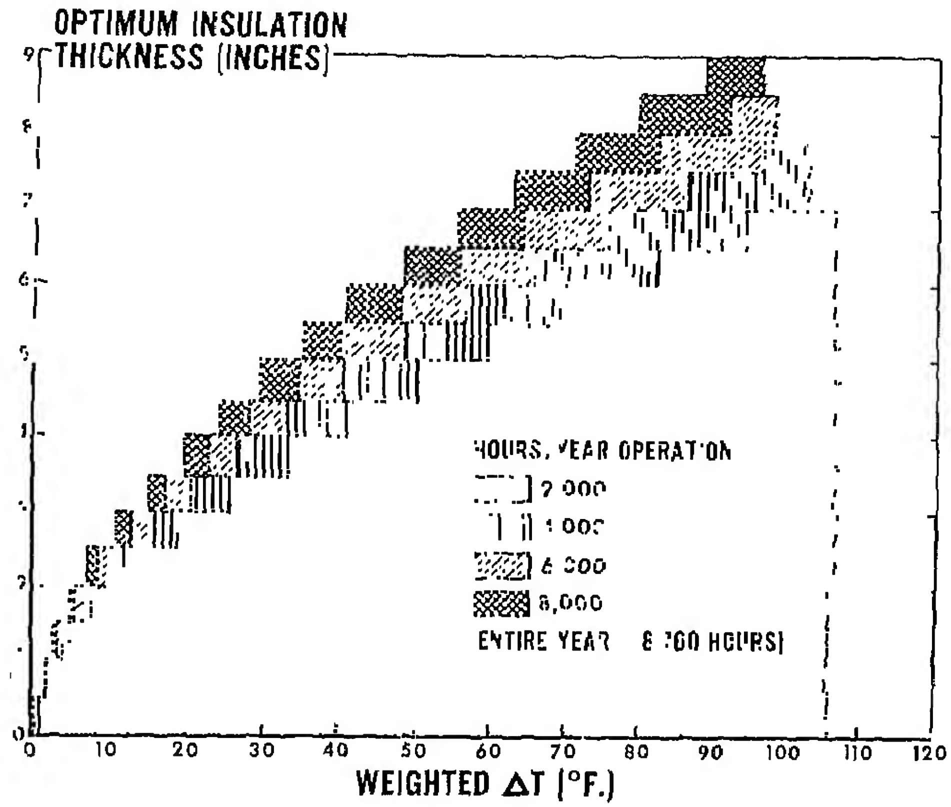


FIGURE 44.—Optimum thickness of expanded-polystyrene insulation for ceilings of rooms above 32° F.—package systems.

(Text continued on page 90.)

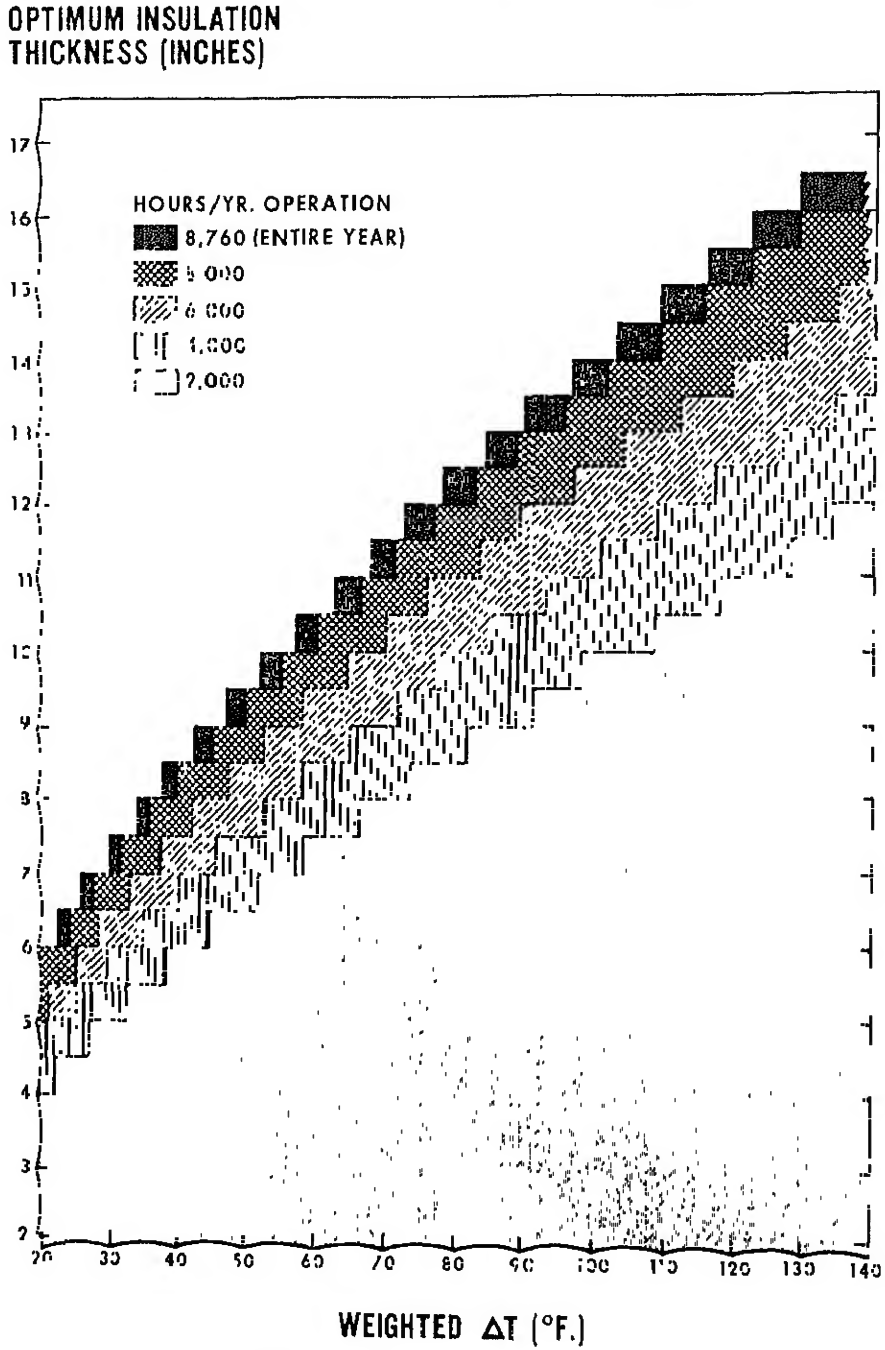


FIGURE 45.—Optimum thickness of fibrous-glass insulation for outside walls and ceilings of rooms 32° F. or lower—package systems.

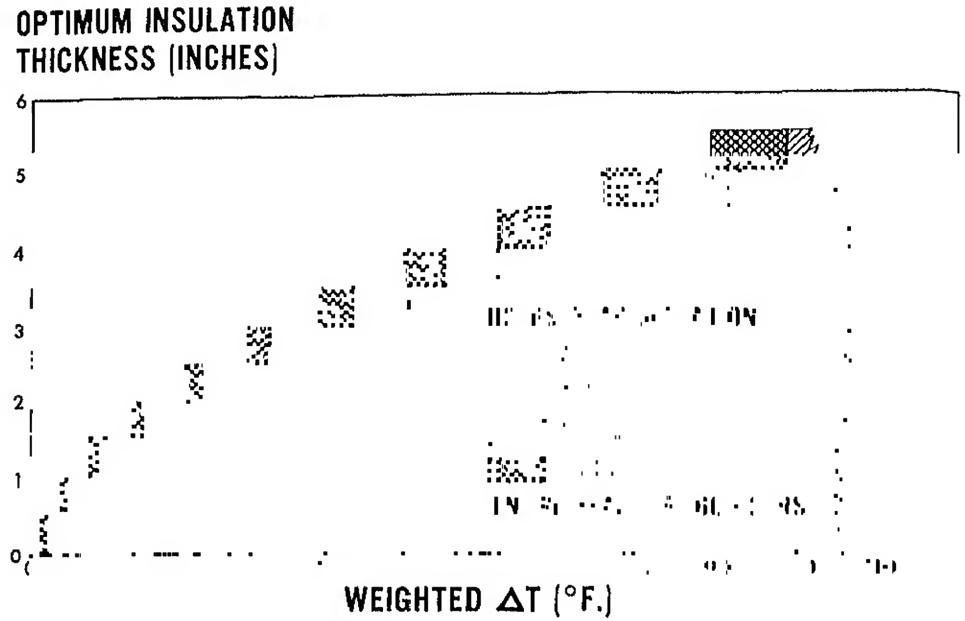


FIGURE 46.—Optimum thickness of expanded-polystyrene insulation for outside walls of rooms above 32° F.—one central system.

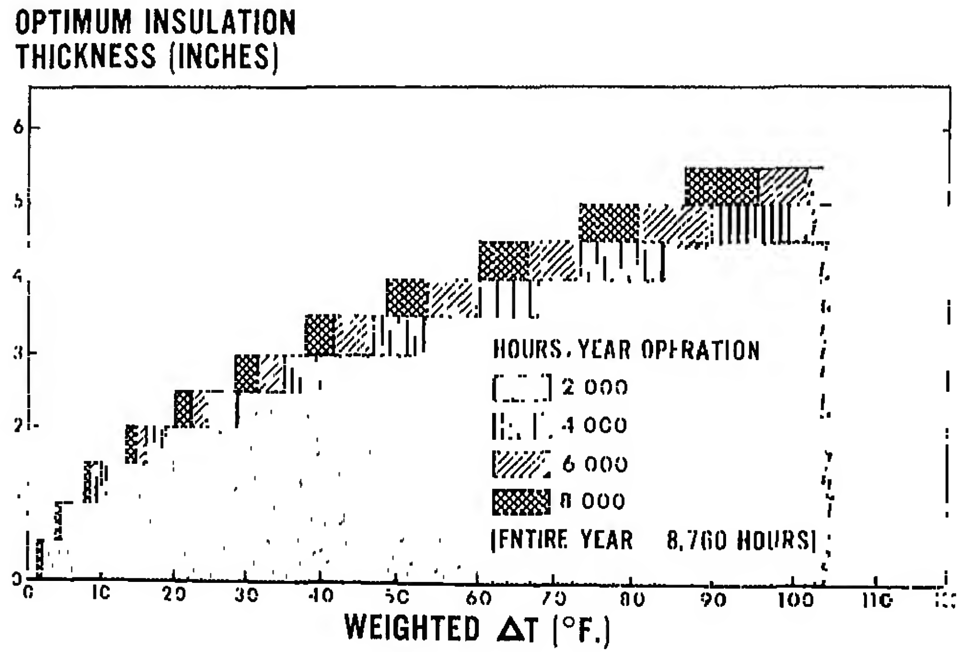


FIGURE 47.—Optimum thickness of expanded-polystyrene insulation for ceilings of rooms above 32° F.—one central system.

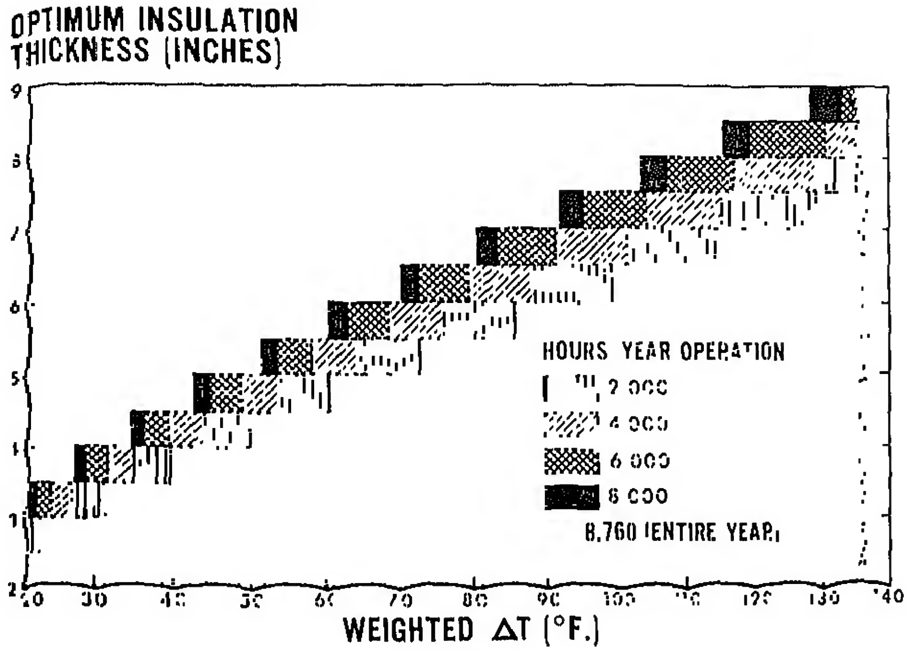


FIGURE 48.—Optimum thickness of fibrous-glass insulation for outside walls and ceilings of rooms 32° F. or lower—one central system.

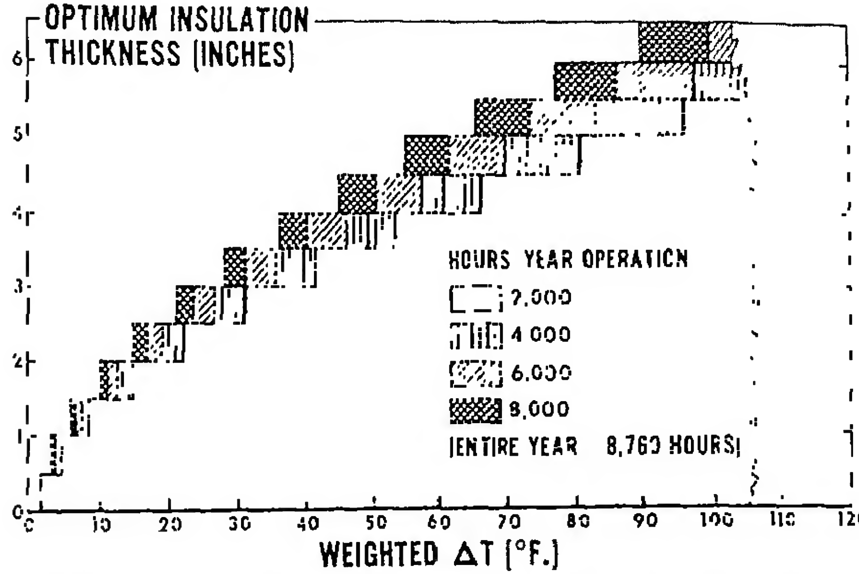


FIGURE 50.—Optimum thickness of expanded-polystyrene insulation for ceilings of rooms above 32° F.—four central systems.

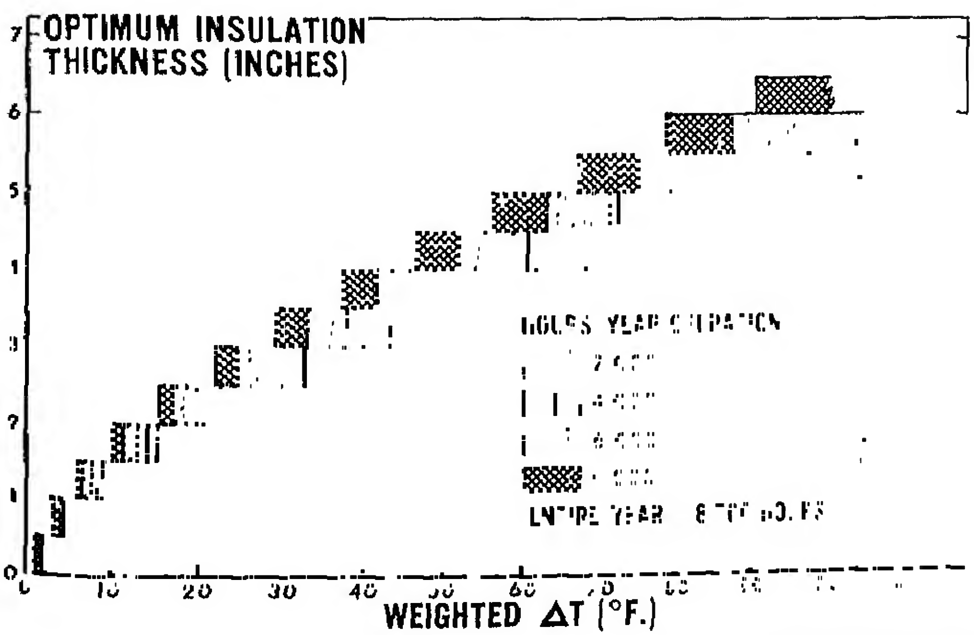


FIGURE 49.—Optimum thickness of expanded-polystyrene insulation for outside walls of rooms above 32° F.—four central systems.

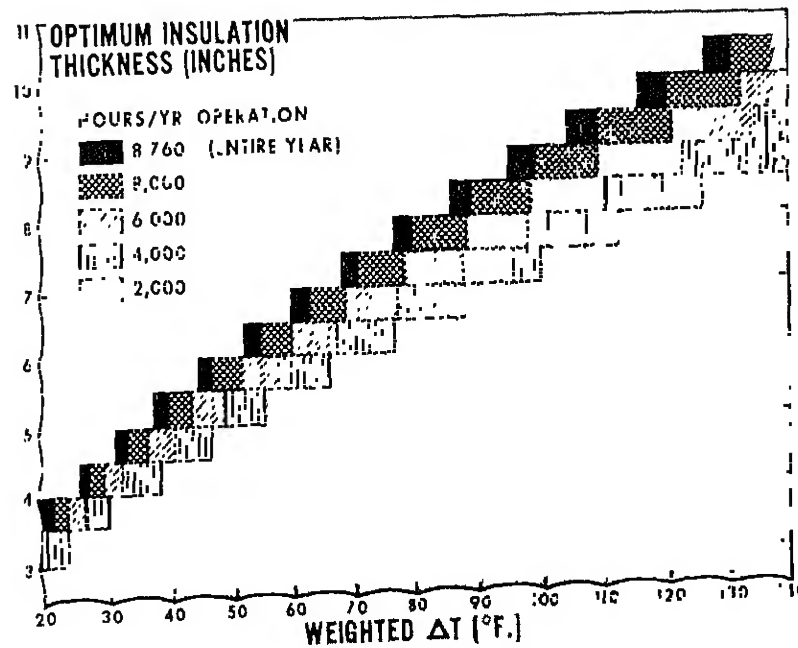


FIGURE 51.—Optimum thickness of fibrous-glass insulation for outside walls and ceilings of rooms 32° F. or lower—four central systems.

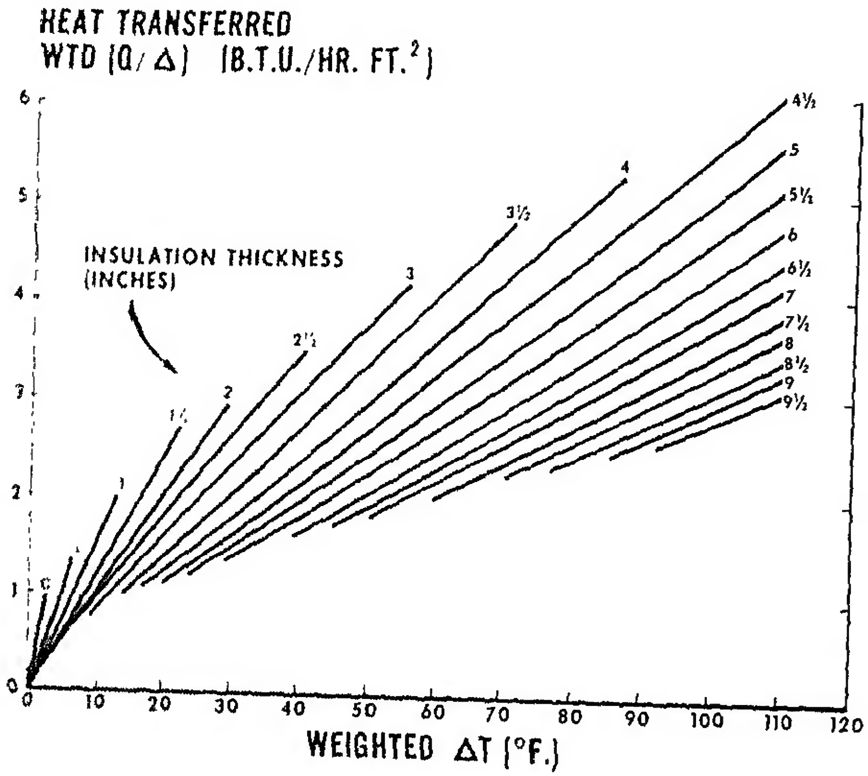


FIGURE 52.—Heat transferred by insulated wall or ceiling in rooms above 32° F., using expanded-polystyrene insulation.

The heat gains shown in these two figures can be used for any type of refrigeration system. They *apply only* if the sum of the thermal resistances used for the wall construction (other than insulation) is reasonably close to that used in the calculations, 2.0 hr.-ft.²-° F./B.t.u.

All of the data obtained from the five preceding steps have been tabulated for 10 different metropolitan areas, Atlanta, Boston, Chicago, Houston, Los Angeles, Orlando, Philadelphia, Phoenix, Seattle, and St. Louis. This information is listed separately for outside walls and ceilings with respect to package systems, four central systems, and one central system (tables 48 through 53). Similar data could be derived for any city by using the tables here. The 10 metropolitan areas selected are representative of climate conditions throughout the United States.

In these tables 48 to 53 the WTD (° F.) and operating hours/year have been calculated from weather data. The value of the optimum insulation thickness was then obtained from the appropriate graph or curve in figures 43 through 51. The heat gain (WTD Q/A) was then obtained from the curves in figures 52 or 53.

With the design information contained in tables 48 through 53 and in figures 43

(Text continued on page 99)

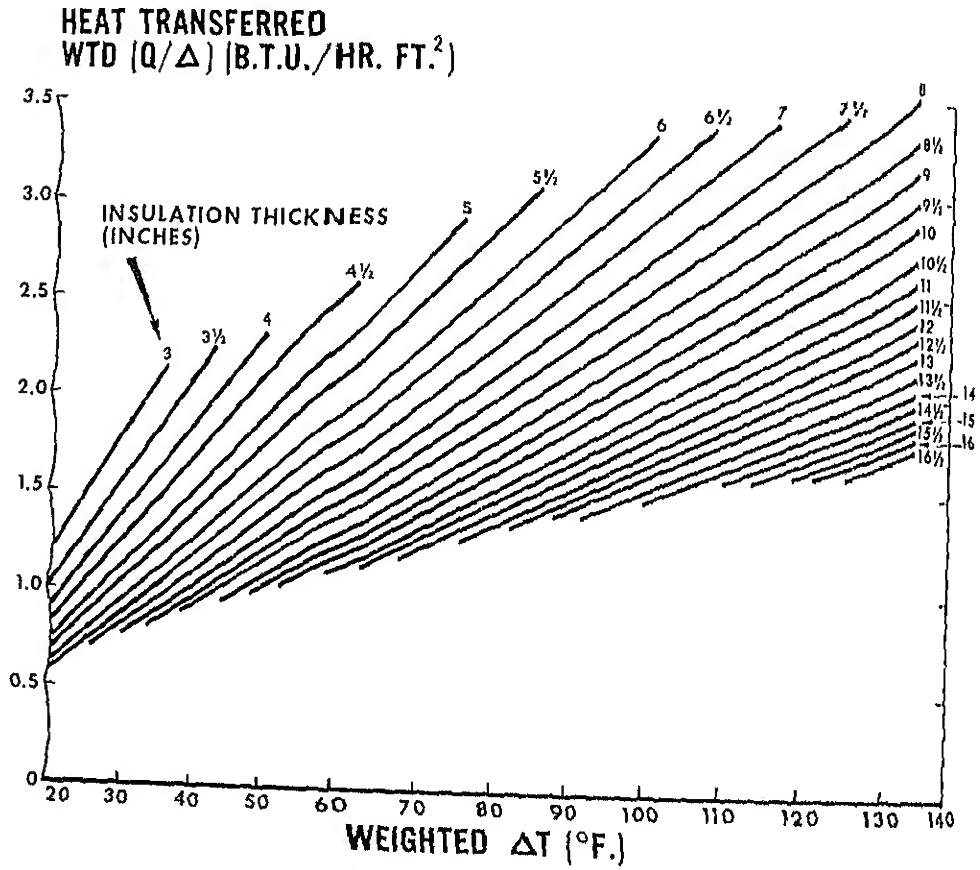


FIGURE 53.—Heat transferred by insulated wall or ceiling in rooms 32° F. or lower, using fibrous-glass insulation.

TABLE 48.—*Data for outside wall calculations for 10 cities, package systems*

Room temperature (° F.)	Type of insulation	Item[1]	Cities									
			Atlanta	Boston	Chicago	Houston	Los Angeles	Orlando	Phila.	Phoenix	Seattle	St. Louis
72	Expanded polystyrene.	WTD[2] ° F.	6.0	5.7	5.7	14.1	4.6	6.7	6.2	14.1	5.4	7.5
		Operating hr./yr	2844.	1381.	1776.	5055.	1056.	5415.	2043.	4089.	660.	2796.
		Optimum thickness inches	1.5	1.0	1.0	2.5	1.0	1.5	1.5	2.5	1.0	1.5
		WTD Q/A B.t.u./hr.-sq.ft.	.78	.98	.98	1.24	.80	.86	.80	1.24	.92	.96
50	Expanded polystyrene.	WTD[2] ° F.	18.4	15.8	17.0	23.2	12.7	22.7	18.0	24.3	11.2	20.3
		Operating hr./yr	6447.	4246.	4629.	7926.	8175.	8442.	4800.	7467.	5316.	5646.
		Optimum thickness inches	3.5	3.0	3.0	4.0	3.0	4.0	3.0	4.0	2.5	3.5
		WTD Q/A B.t.u./hr.-sq.ft.	1.22	1.20	1.29	1.36	.96	1.34	1.36	1.42	.98	1.32
45	Expanded polystyrene.	WTD[2] ° F.	23.4	20.8	22.0	28.2	17.7	27.7	23.0	29.3	16.2	25.0
		Operating hr./yr	6447.	4246.	4629.	7926.	8175.	8442.	4800.	7467.	5316.	5646.
		Optimum thickness inches	4.0	3.5	3.5	4.5	3.5	4.5	3.5	4.5	3.0	4.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.37	1.38	1.46	1.50	1.17	1.46	1.52	1.56	1.13	1.32
40	Expanded polystyrene.	WTD[2] ° F.	24.7	20.5	22.3	31.2	21.5	31.9	22.1	30.6	16.3	26.2
		Operating hr./yr	7665.	5713.	5886.	8553.	8757.	8700.	6444.	8562.	7623.	6660.
		Optimum thickness inches	4.0	3.5	3.5	5.0	4.0	5.0	3.5	5.0	3.5	4.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.44	1.35	1.47	1.52	1.26	1.55	1.47	1.50	1.08	1.54
[3]32	Fibrous glass.	WTD[2] ° F.	29.9	21.9	24.7	38.4	29.5	39.7	25.2	37.8	21.6	30.1
		Operating hr./yr	8469.	7693.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	7.0	6.0	6.0	8.5	7.0	8.5	6.5	8.0	6.0	7.0
		WTD Q/A B.t.u./hr.-sq.ft.	.84	.71	.80	.91	.83	.94	.76	.95	.70	.85
[3]25	Fibrous glass.	WTD[2] ° F.	36.9	28.9	31.7	45.4	36.5	46.7	32.2	44.8	28.6	37.1
		Operating hr./yr	8469.	7693.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	8.0	6.5	7.0	9.0	8.0	9.0	7.0	9.0	7.0	7.5
		WTD Q/A B.t.u./hr.-sq.ft.	.92	.87	.89	1.02	.92	1.05	.91	1.01	.80	.98
−10	Fibrous glass.	WTD[2] ° F.	70.6	59.9	60.9	80.3	71.5	81.7	63.9	79.8	63.5	66.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	11.5	10.5	10.5	12.5	11.5	12.5	11.0	12.5	11.0	11.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.29	1.18	1.20	1.37	1.30	1.40	1.20	1.36	1.20	1.27
−20	Fibrous glass.	WTD[2] ° F.	80.6	69.9	70.9	90.3	81.5	91.7	73.6	89.8	73.1	76.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	12.5	11.5	11.5	13.0	12.5	13.5	12.0	13.0	12.0	12.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.38	1.28	1.29	1.50	1.40	1.46	1.29	1.49	1.28	1.35

[1] Q/A values and insulation thicknesses for cities included above were derived from the curves in figures 43 through 53. Because readings from a curve vary slightly, the figures for Chicago will not agree exactly with those given in table 3, which were taken directly from computer runs.

[2] Weighted temperature difference.

[3] Fibrous glass and expanded polystyrene may be interchangeable for this temperature.

TABLE 49.—*Data for ceiling calculations for 10 cities, package systems*

Room temperature (° F.)	Type of insulation	Item[1]	Cities: Atlanta	Boston	Chicago	Houston	Los Angeles	Orlando	Phila.	Phoenix	Seattle	St Louis
72	Expanded polystyrene.	WTD[2] ° F.	36.6	32.9	35.0	34.5	39.6	35.4	34.6	43.8	35.0	36.7
		Operating hr./yr	4410.	3658.	3736.	5735.	3688.	6075.	3834.	5275.	3196.	4250.
		Optimum thickness inches	4.5	4.0	4.5	4.5	4.5	5.0	4.5	5.5	4.5	4.5
		WTD Q/A B.t.u./hr.-sq.ft.	1.97	1.94	1.87	1.84	2.14	1.74	1.85	1.98	1.87	[illegible]
50	Expanded polystyrene.	WTD ° F.	43.3	40.0	41.9	43.7	32.8	44.4	42.0	49.1	34.9	44.7
		Operating hr./yr	6500.	5450.	5539.	7926.	8175.	8442.	5643.	7467.	5769.	5985.
		Optimum thickness inches	5.5	5.0	5.0	6.0	5.0	6.0	5.0	6.0	5.0	5.5
		WTD Q/A B.t.u./hr.-sq.ft.	1.95	1.97	2.07	1.82	1.61	1.84	2.07	2.07	1.70	2.01
45	Expanded polystyrene.	WTD ° F.	48.6	45.0	46.9	48.7	37.8	49.4	47.0	54.1	39.9	49.7
		Operating hr./yr	6500.	5450.	5539.	7926.	8175.	8442.	5643.	7467.	5769.	5985.
		Optimum thickness inches	6.0	5.5	5.5	6.0	5.5	6.5	5.5	6.5	5.0	6.0
		WTD Q/A B.t.u./hr.-sq.ft.	2.04	2.03	2.11	2.04	1.70	1.94	2.12	2.12	1.96	2.08
40	Expanded polystyrene.	WTD ° F.	46.2	44.6	46.1	50.2	40.3	53.0	45.1	52.2	35.2	49.2
		Operating hr./yr	6775.	6198.	6322.	8553.	8757.	8700.	6623.	8562.	7623.	6734.
		Optimum thickness inches	6.0	5.5	5.5	6.5	6.0	6.5	5.5	6.5	5.0	6.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.93	2.01	2.09	1.96	1.68	2.08	2.03	2.04	1.72	2.06
[3]32	Fibrous glass.	WTD ° F.	49.3	43.0	46.5	56.9	48.3	60.6	45.1	58.9	38.0	50.6
		Operating hr./yr	8479.	7683.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	9.5	8.0	8.5	10.0	9.5	10.5	8.5	10.5	8.0	9.0
		WTD Q/A B.t.u./hr.-sq ft.	1.06	1.08	1.11	1.14	1.04	1.20	1.08	1.16	.95	1.14
[3]25	Fibrous glass.	WTD ° F.	56.3	50.0	53.5	63.9	55.3	67.6	52.1	65.9	45.0	67.6
		Operating hr./yr	8479.	7683.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	10.0	9.0	9.0	11.0	10.0	11.5	9.5	11.0	9.0	10.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.16	1.13	1.21	1.21	1.14	1.23	1.12	1.24	1.01	1.19
−10	Fibrous glass.	WTD ° F.	89.4	78.4	79.4	98.8	90.3	102.6	81.5	100.9	79.9	84.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	13.0	12.0	12.5	14.0	13.0	14.0	12.5	14.0	12.5	12.5
		WTD Q/A B.t.u./hr.-sq.ft.	1.48	1.38	1.35	1.53	1.50	1.60	1.39	1.56	1.36	1.45
−20	Fibrous glass.	WTD ° F.	99.4	88.4	89.4	108.8	100.3	112.6	91.5	110.9	89.9	94.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	14.0	13.0	13.0	14.5	14.0	15.0	13.5	15.0	13.0	13.5
		WTD Q/A B.t.u./hr.-sq.ft.	1.54	1.46	1.48	1.65	1.56	1.66	1.46	1.62	1.49	1.59

[1] Q/A values and insulation thicknesses for cities included above were derived from the curves in figures 43 through 53. Because readings from a curve vary slightly, the figures for Chicago will not agree exactly with those given in table 4, which were taken directly from computer runs.

[2] Weighted temperature difference.

[3] Fibrous glass and expanded polystyrene may be interchangeable for this temperature.

TABLE 50.—*Data for outside wall calculations for 10 cities, one central system*

Room temperature (° F.)	Type of insulation	Item[1]	Cities: Atlanta	Boston	Chicago	Houston	Los Angeles	Orlando	Phila.	Phoenix	Seattle	St. Louis
72	Expanded polystyrene.	WTD[2] ° F.	6.0	5.7	5.7	14.1	4.6	6.7	6.2	14.1	5.4	7.5
		Operating hr./yr	2844.	1381.	1776.	5055.	1056.	5415.	2043.	4089.	660.	2796.
		Optimum thickness inches	1.0	.5	.5	1.5	.5	1.0	1.0	1.5	.5	1.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.05	1.42	1.42	1.86	1.18	1.16	1.08	1.86	1.38	1.30
50	Expanded polystyrene.	WTD[2] ° F.	18.4	15.8	17.0	23.2	12.7	22.7	18.0	24.3	11.2	20.0
		Operating hr./yr	6447.	4246.	4629.	7926.	8175.	8442.	4800.	7467.	5316.	5646.
		Optimum thickness inches	2.0	1.5	1.5	2.5	1.5	2.5	2.0	2.5	1.5	2.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.91	2.07	2.23	2.05	1.65	2.00	1.88	2.15	1.46	2.10
45	Expanded polystyrene.	WTD[2] ° F.	23.4	20.8	22.0	28.2	17.7	27.7	23.0	29.3	16.2	25.0
		Operating hr./yr	6447.	4246.	4629.	7926.	8175.	8442.	4800.	7467.	5316.	5646.
		Optimum thickness inches	2.5	2.0	2.0	2.5	2.0	2.5	2.0	2.5	1.5	2.5
		WTD Q/A B.t.u./hr.-sq.ft.	2.06	2.19	2.33	2.49	1.85	2.46	2.43	2.60	2.13	2.22
40	Expanded polystyrene.	WTD[2] ° F.	24.7	20.5	22.3	21.2	21.5	31.9	22.1	30.6	16.3	26.2
		Operating hr./yr	7665.	5713.	5886.	8553.	8757.	8700.	6444.	8562.	7623.	6660.
		Optimum thickness inches	2.5	2.0	2.0	3.0	2.5	3.0	2.0	3.0	2.0	2.5
		WTD Q/A B.t.u./hr.-sq.ft.	2.19	2.14	2.36	2.40	1.90	2.43	2.33	2.35	1.70	2.32
[3]32	Fibrous glass.	WTD[2] ° F.	29.9	21.9	24.7	38.4	29.5	39.7	25.2	37.8	21.6	30.1
		Operating hr./yr	8469.	7693.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	4.0	3.0	3.5	4.5	4.0	4.5	3.5	4.5	3.5	4.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.41	1.33	1.32	1.69	1.40	1.70	1.34	1.62	1.15	1.43
[3]25	Fibrous glass.	WTD[2] ° F.	36.9	28.9	31.7	45.4	36.5	46.7	32.2	44.8	28.6	37.1
		Operating hr./yr	8469.	7693.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	4.5	3.5	4.0	5.0	4.5	5.0	4.0	5.0	4.0	4.5
		WTD Q/A B.t.u./hr.-sq.ft.	1.58	1.54	1.50	1.78	1.56	1.83	1.53	1.76	1.35	1.59
−10	Fibrous glass.	WTD[2] ° F.	70.6	59.9	60.9	80.3	71.5	81.7	63.6	79.8	63.5	66.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	6.5	6.0	6.0	7.0	6.5	7.0	6.0	7.0	6.0	6.0
		WTD Q/A B.t.u./hr.-sq.ft.	2.22	2.01	2.04	2.38	2.25	2.43	2.14	2.36	2.14	2.26
−20	Fibrous glass.	WTD[2] ° F.	80.6	69.9	70.9	90.3	81.5	91.7	73.6	89.8	73.1	76.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	7.0	6.5	6.5	7.0	7.0	7.5	6.5	7.0	6.5	6.5
		WTD Q/A B.t.u./hr.-sq.ft.	2.39	2.20	2.23	2.71	2.42	2.57	2.32	2.60	2.30	2.43

[1] Q/A values and insulation thicknesses for cities included above were derived from the curves in figures 43 through 53. Because readings from a curve vary slightly, the figures for Chicago will not agree exactly with those given in table 5, which were taken directly from computer runs.

[2] Weighted temperature difference.

[3] Fibrous glass and expanded polystyrene may be interchangeable for this temperature.

TABLE 51.—*Data for ceiling calculations for 10 cities, one central system*

Room temperature (° F.)	Type of insulation	Item[1]	Cities: Atlanta	Boston	Chicago	Houston	Los Angeles	Orlando	Phila.	Phoenix	Seattle	St. Louis
72	Expanded polystyrene	WTD[2] ° F.	36.6	32.9	35.0	34.5	39.6	35.4	34.6	43.8	35.0	36.7
		Operating hr./yr	4410.	3658.	3736.	5735.	3688.	6075.	3834.	5275.	3196.	4250.
		Optimum thickness inches	3.0	2.5	2.5	3.0	3.0	3.0	2.5	3.5	2.5	3.0
		WTD Q/A B.t.u./hr.-sq.ft.	2.82	2.93	3.14	2.65	3.05	2.71	3.10	2.96	3.14	2.82
50	Expanded polystyrene.	WTD[2] ° F.	43.3	40.0	41.9	43.7	32.8	44.4	42.0	49.1	34.9	44.7
		Operating hr./yr	6500.	5450.	5539.	7926.	8175.	8442.	5643.	7467.	5769.	5985.
		Optimum thickness inches	3.5	3.0	3.0	3.5	3.0	3.5	3.0	3.5	3.0	3.5
		WTD Q/A B.t.u./hr.-sq.ft.	2.93	3.08	3.22	2.95	2.52	3.01	3.24	3.15	2.68	3.04
45	Expanded polystyrene	WTD[2] ° F.	48.6	45.0	46.9	48.7	37.8	49.4	47.0	54.1	39.9	49.7
		Operating hr./yr	6500.	5450.	5539.	7926.	8175.	8442.	5643.	7467.	5769.	5985.
		Optimum thickness inches	3.5	3.5	3.5	3.5	3.5	4.0	3.5	4.0	3.0	3.5
		WTD Q/A B.t.u./hr.-sq.ft.	3.31	3.06	3.19	3.32	2.55	2.98	3.19	3.30	3.07	3.39
40	Expanded polystyrene.	WTD[2] ° F.	46.2	44.6	46.1	50.2	40.3	53.0	45.1	52.2	35.2	49.2
		Operating hr./yr	7675.	6198.	6322.	8553.	8757.	8700.	6623.	8562.	7023.	6734.
		Optimum thickness inches	3.5	3.5	3.5	4.0	3.5	4.0	3.5	4.0	3.0	3.5
		WTD Q/A B.t.u./hr.-sq.ft.	3.13	3.03	3.14	3.05	2.72	3.23	3.07	3.18	2.71	3.35
[3]32	Fibrous glass.	WTD[2] ° F.	49.3	43.0	46.5	56.9	48.3	60.6	45.1	58.9	38.0	50.6
		Operating hr./yr	8479.	7683.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	5.0	4.5	5.0	5.5	5.0	6.0	5.0	5.5	4.5	5.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.95	1.85	1.82	2.07	1.90	2.04	1.77	2.15	1.63	2.00
[3]25	Fibrous glass.	WTD[2] ° F.	56.3	50.0	53.5	63.9	55.3	67.6	52.1	65.9	45.0	57.6
		Operating hr./yr	8479.	7683.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	5.5	5.0	5.0	6.0	5.5	6.0	5.0	6.0	5.0	5.5
		WTD Q/A B.t.u./hr.-sq.ft.	2.04	1.97	2.11	2.15	2.00	2.28	2.06	2.22	1.76	2.10
−10	Fibrous glass.	WTD[2] ° F.	89.4	78.4	79.4	98.8	90.3	102.6	81.5	100.9	79.9	84.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	7.0	6.5	6.5	7.5	7.0	8.0	7.0	7.5	7.0	7.0
		WTD Q/A B.t.u./hr.-sq.ft	2.67	2.48	2.52	2.79	2.71	2.75	2.42	2.86	2.36	2.63
−20	Fibrous glass.	WTD[2] ° F.	99.4	88.4	98.4	108.8	100.3	112.6	91.5	110.9	89.9	94.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	7.5	7.0	7.0	8.0	7.5	8.0	7.5	8.0	7.0	7.5
		WTD Q/A B.t.u./hr.-sq.ft.	2.81	2.64	2.67	2.92	2.84	3.04	2.56	2.99	2.70	2.60

[1] Q/A values and insulation thicknesses for cities included above were derived from the curves in figures 43 through 53. Because readings from a curve vary slightly, the figures for Chicago will not agree exactly with those given in table 6, which were taken directly from computer runs.

[2] Weighted temperature difference.

[3] Fibrous glass and expanded polystyrene may be interchangeable for this temperature.

TABLE 52.—*Data for outside wall calculations for 10 cities, four central systems*

Room temperature (° F.)	Type of insulation	Item[1]	Cities									
			Atlanta	Boston	Chicago	Houston	Los Angeles	Orlando	Phila.	Phoenix	Seattle	St. Louis
72	Expanded polystyrene.	WTD[2] ° F	6.0	5.7	5.7	14.1	4.6	6.7	6.2	14.1	5.4	7.5
		Operating hr./yr	2844.	1381.	1776.	5055.	1056.	5415.	2043.	4089.	660.	2796.
		Optimum thickness inches	1.0	1.0	1.0	2.0	.5	1.0	1.0	2.0	1.0	1.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.04	.98	1.00	1.47	1.16	1.16	1.08	1.47	.92	1.30
50	Expanded polystyrene.	WTD[2] ° F.	18.4	15.8	17.0	23.2	12.7	22.7	18.0	24.3	11.2	20.0
		Operating hr./yr	6447.	4246.	4629.	7926.	8175.	8442.	4800.	7467.	5316.	5646.
		Optimum thickness inches	2.5	2.0	2.0	3.0	2.0	3.0	2.0	3.0	1.5	2.5
		WTD Q/A B.t.u /hr.-sq.ft.	1.61	1.65	1.73	1.77	1.32	1.72	1.93	1.86	1.46	1.77
45	Expanded polystyrene.	WTD[2] ° F	23.4	20.8	22.0	28.2	17.7	27.7	23.0	29.3	16.2	25.0
		Operating hr./yr	6447.	4246.	4629.	7926.	8175.	8442.	4800.	7467.	5316.	5646.
		Optimum thickness inches	2.5	2.5	2.5	3.0	2.5	3.0	2.5	3.0	2.0	2.5
		WTD Q/A B t.u./hr.-sq ft.	2.07	1.83	1.94	2.15	1.56	2.11	2.04	2.24	1.68	2.21
40	Expanded polystyrene.	WTD[2] ° F.	24.7	20.5	22.3	31.2	21.5	31.9	22.1	30.6	16.3	26.2
		Operating hr./yr	7665.	5714.	5886.	8553.	8757.	8700.	6444.	8562.	7623.	6660.
		Optimum thickness inches	3.0	2.5	2.5	3.5	3.0	3.5	2.5	3.5	2.5	3.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.87	1.82	1.96	2.08	1.64	2.13	1.96	2.05	1.43	2.00
[3]32	Fibrous glass.	WTD[2] ° F.	29.9	21.9	24.7	38.4	29.5	39.7	25.2	37.8	21.6	30.1
		Operating hr./yr	8469.	7693.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	4.5	4.0	4.0	5.5	4.5	5.5	4.0	5.5	4.0	4.5
		WTD Q/A B.t.u./hr.-sq.ft.	1.27	1.03	1.16	1.36	1.25	1.41	1.18	1.34	1.01	1.28
[3]25	Fibrous glass.	WTD[2] ° F.	36.9	28.9	31.7	45.4	36.5	46.7	32.2	44.8	28.6	37.1
		Operating hr./yr	8469.	7693.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7668.
		Optimum thickness inches	5.0	4.5	4.5	6.0	5.0	6.0	4.5	6.0	4.5	5.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.44	1.22	1.35	1.50	1.42	1.54	1.37	1.48	1.22	1.45
−10	Fibrous glass.	WTD[2] ° F.	70.6	59.9	60.9	80.3	71.5	81.7	63.6	79.8	63.5	66.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	7.5	7.0	7.0	8.0	7.5	8.0	7.0	8.0	7.0	7.0
		WTD Q/A B.t.u./hr.-sq.ft.	1.94	1.74	1.78	2.11	1.97	2.14	1.86	2.09	1.86	1.96
−20	Fibrous glass.	WTD[2] ° F.	80.6	69.9	70.9	90.3	81.5	91.7	73.6	89.8	73.1	76.9
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	8.0	7.5	7.5	8.5	8.0	8.5	7.5	8.5	7.5	8.0
		WTD Q/A B.t.u./hr.-sq ft.	2.11	1.92	1.94	2.25	2.14	2.29	2.03	2.24	2.02	2.01

[1] Q/A values and insulation thicknesses for cities included above were derived from the curves in figures 43 through 53. Because readings from a curve vary slightly, the figures for Chicago will not agree exactly with those given in table 7, which were taken directly from computer runs.

[2] Weighted temperature difference.

[3] Fibrous glass and expanded polystyrene may be interchangeable for this temperature.

TABLE 53.—*Data for ceiling calculations for 10 cities, four central systems*

Room temperature (° F.)	Type of insulation	Item[1]	Cities: Atlanta	Boston	Chicago	Houston	Los Angeles	Orlando	Phila.	Phoenix	Seattle	St. L
72	Expanded polystyrene.	WTD[2] ° F.	36.6	32.9	35.0	34.5	39.6	35.4	34.6	43.8	35.0	30
		Operating hr./yr	4410.	3658.	3736.	5735.	3688.	6075.	3834.	5275.	3196.	4250
		Optimum thickness inches	3.5	3.0	3.0	3.5	3.5	3.5	3.0	4.0	3.0	3
		WTD Q/A B.t.u./hr.-sq.ft.	2.48	2.52	2.70	2.31	2.67	2.38	2.66	2.63	2.70	2
50	Expanded polystyrene.	WTD[2] ° F.	43.3	40.0	41.9	43.7	32.8	44.4	42.0	49.1	34.9	44
		Operating hr./yr	6500.	5450.	5539.	7926.	8175.	8442.	5643.	7467.	5769.	5985
		Optimum thickness inches	4.0	3.5	4.0	4.0	3.5	4.5	4.0	4.5	3.5	4
		WTD Q/A B.t.u./hr.-sq.ft.	2.60	2.70	2.51	2.62	2.19	2.41	2.53	2.68	2.34	2
45	Expanded polystyrene.	WTD[2] ° F.	48.6	45.0	46.9	48.7	37.8	49.4	47.0	54.1	39.9	40
		Operating hr./yr	6500.	5450.	5539.	7926.	8175.	8442.	5643.	7467.	5769.	5985
		Optimum thickness inches	4.0	4.0	4.0	4.5	4.0	4.5	4.0	4.5	3.5	4
		WTD Q/A B.t.u./hr.-sq.ft.	2.93	2.71	2.82	2.66	2.26	2.70	2.83	2.98	2.64	3
40	Expanded polystyrene.	WTD[2] ° F.	46.2	44.6	46.1	50.2	40.3	53.0	45.1	52.2	35.2	49
		Operating hr./yr	7675.	6198.	6322.	8553.	8757.	8700.	6623.	8562.	7023.	6734
		Optimum thickness inches	4.5	4.0	4.0	4.5	4.0	5.0	4.0	4.5	3.5	4
		WTD Q/A B.t.u./hr.-sq.ft.	2.51	2.68	2.78	2.75	2.42	2.64	2.72	2.86	2.36	2
[3]32	Fibrous glass.	WTD[2] ° F.	49.3	43.0	46.5	56.9	48.3	60.6	45.1	58.9	38.0	50
		Operating hr./yr	8479.	7683.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7068.
		Optimum thickness inches	6.0	5.5	5.5	6.5	6.0	7.0	5.5	6.5	5.5	6.
		WTD Q/A B.t.u./hr.-sq.ft.	1.64	1.54	1.67	1.77	1.60	1.77	1.62	1.84	1.35	1.
[3]25	Fibrous glass.	WTD[2] ° F.	56.3	50.0	53.5	63.9	55.3	67.6	52.1	65.9	45.0	57.
		Operating hr./yr	8479.	7683.	7422.	8745.	8760.	8760.	7881.	8760.	8742.	7068.
		Optimum thickness inches	6.5	6.0	6.0	7.0	6.5	7.5	6.0	7.0	6.0	6.
		WTD Q/A B.t.u./hr.-sq.ft.	1.75	1.66	1.78	1.87	1.72	1.85	1.74	1.93	1.48	1.
−10	Fibrous glass.	WTD[2] ° F.	89.4	78.4	79.4	98.8	90.3	102.6	81.5	100.9	79.9	84.
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	8.5	8.0	8.0	9.0	8.5	9.0	8.0	9.0	8.0	8.
		WTD Q/A B.t.u./hr.-sq.ft.	2.24	2.05	2.08	2.35	2.25	2.45	2.14	2.40	2.09	2.
−20	Fibrous glass.	WTD[2] ° F.	99.4	88.4	89.4	108.8	100.3	112.6	91.5	110.9	89.9	94.
		Operating hr./yr	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.	8760.
		Optimum thickness inches	9.0	8.5	8.5	9.5	9.0	9.5	8.5	9.5	8.5	9.
		WTD Q/A B.t.u./hr.-sq.ft.	2.37	2.20	2.21	2.48	2.39	2.58	2.28	2.54	2.12	2.

[1] Q/A values and insulation thicknesses for cities included above were derived from the curves in figures 43 through 53. Because readings from a curve vary slightly, the figures for Chicago will not agree exactly with those given in table 8, which were taken directly from computer runs.

[2] Weighted temperature difference.

[3] Fibrous glass and expanded polystyrene may be interchangeable for this temperature.

through 53, it is possible to calculate the cost differentials caused by the differences in climate between Chicago and other cities. The calculations of these cost differentials are illustrated in the following paragraphs. It is assumed that the central food distribution center, using one central refrigeration system, is relocated f Chicago, Ill., to Orlando, Fla.

Worksheets for Orlando and Chicago are illustrated in tables 54 and 55.

TABLE 54.—*Insulation requirements and heat gains, Chicago food distribution center*

Room temperature (° F.)	Type insulation	Surface area (ft.²)	Insulation: Thickness (in.)	Insulation: Board ft. required	Heat gain: WTD Q/A (B.t.u./hr.-ft.²)	Heat gain: Tons refrigeration	Heat gain: Operating days per year	Heat gain: Ton-days
Ceiling:								
72	Expanded polystyrene	6,270	2.5	15,675	3.14	1.641	156	256
50	do	27,551	3.0	82,653	3.22	7.393	231	1,708
45	do	1,610	3.5	5,635	3.19	.428	231	99
40	do	6,806	3.5	23,818	3.14	1.781	263	468
32	Fibrous glass	32,731	5.0	163,655	1.82	4.964	309	1,534
25	do	1,750	5.0	8,750	2.11	.308	309	95
−10	do	9,884	6.5	64,246	2.52	2.076	365	758
−20	do	288	7.0	2,016	2.67	.064	365	23
Subtotals	Expanded polystyrene			127,781		18.655		4,941
	Fibrous glass			238,667				
Outside wall:								
72	Expanded polystyrene	4,499	0.5	2,250	1.42	.532	74	39
50	do	18,379	1.5	27,569	2.23	3.415	193	659
45	do	966	2.0	1,932	2.33	.188	193	36
40	do	5,055	2.0	10,110	2.36	.994	245	244
32	Fibrous glass	13,326	3.5	46,641	1.32	1.465	309	453
25	do	1,030	4.0	4,120	1.50	.129	309	40
−10	do	4,724	6.0	28,344	2.04	.803	365	293
−20	do							
Subtotals	Expanded polystyrene			41,861		7.526		1,764
	Fibrous glass			79,105				
Totals	Expanded polystyrene			169,642		26.18		6,705
	Fibrous glass			317,772				

TABLE 55.—*Insulation requirements and heat gains, Orlando food distribution center*

Room temperature (° F.)	Type insulation	Surface area (ft.²)	Insulation: Thickness (in.)	Insulation: Board ft. required	Heat gain: WTD Q/A (B.t.u./hr.-ft.²)	Heat gain: Tons refrigeration	Heat gain: Operating days per year	Heat gain: Ton-days
Ceiling:								
72	Expanded polystyrene	6,270	3.0	18,810	2.71	1.415	226	358
50	do	27,551	3.5	96,429	3.01	6.910	352	2,432
45	do	1,610	4.0	6,440	2.98	.400	352	141
40	do	6,806	4.0	27,224	3.23	1.832	363	665
32	Fibrous glass	32,731	6.0	196,386	2.04	5.564	365	2,031
25	do	1,750	6.0	10,500	2.28	.332	365	121
−10	do	9,884	8.0	79,702	2.75	2.265	365	827
−20	do	288	8.0	2,304	3.04	.073	365	27
Subtotals	Expanded polystyrene			148,903		18.791		6,602
	Fibrous glass			288,892				
Outside wall:								
72	Expanded polystyrene	4,499	1.0	4,499	1.16	.435	226	98
50	do	18,379	2.5	45,948	2.00	3.063	352	1,078
45	do	966	2.5	2,415	2.46	.198	352	70
40	do	5,055	3.0	15,165	2.43	1.023	363	371
32	Fibrous glass	13,326	4.5	59,967	1.70	1.887	365	689
25	do	1,030	5.0	5,150	1.83	.159	365	58
−10	do	4,724	7.0	33,068	2.43	.956	365	349
−20	do							
Subtotals	Expanded polystyrene			68,027		7.719		2,713
	Fibrous glass			98,185				
Total	Expanded polystyrene			216,930		26.51		9,315
	Fibrous glass			387,077				

1. The surface areas are based on the square feet of insulation required for all four buildings (outside walls and ceilings only, since the inside walls and floors would require the same type and amount of insulation).

2. The number of board feet of insulation required is calculated by multiplying the area (ft.^{2}) by the thickness (inches).

3. The heat gain per square foot (WTD Q/A) is obtained from figures 52 and 53. The tons of refrigeration required because of heat gain are found by multiplying the area (ft.^{2}) by WTD Q/A (B.t.u./ hr.–ft.^{2}), and dividing by 12,000 B.t.u./hr./TR.

4. The operating days per year are determined by dividing the operating hours/year, listed in tables 50 and 51, by 24 hours per day.

5. The ton-days are calculated by multiplying the tons of refrigeration by the operating days per year.

6. The totals of the columns in the worksheets are applied to the cost figures developed in "Situation II, One Central System for Four Buildings," and listed in "Cost Comparisons for the Three Situations," to determine the difference in initial capital expenditures and in owning and operating costs between Chicago and Orlando.

The difference in initial capital expenditures is calculated as follows:

(1) Expanded-polystyrene insulation required for ceilings and outside walls:
Orlando = 216,930 bd. ft.
Chicago = 169,642 bd. ft.
\+ 47,288 bd. ft.
Additional cost 47,288 × \$0.085 = \$4,020

(2) Fibrous-glass insulation required for ceilings and outside walls:
Orlando = 387,077 bd. ft.
Chicago = 317,772 bd. ft.
\+ 69,305 bd. ft.
Additional cost = 69,305 × \$0.13/bd. ft. = \$9,010

(3) Additional refrigeration-system cost due to heat gains:
Orlando = 26.51 tons
Chicago = 26.18 tons
\+ 0.33 tons
Additional cost = 0.33 × \$1,189 = \$392

Total difference in initial capital expenditures = + \$13,422

The difference in refrigeration operating costs caused by differences in *cli*
is obtained from the difference in "ton-days" as follows:

Orlando = 9,315 ton-days
Chicago = 6,705 ton-days
\+ 2,610 ton-days

Additional cost = 2,610 × \$0.463 = \$1,208/year. This difference in opera
costs is added to the base operating cost at Chicago before making any correc
for differences in electric power rates. See "Electric Energy Costs."

Similar cost differentials can be established between Chicago and any other
by using the material presented in this report.

TYPICAL SPECIFICATIONS

Typical specifications have been written for the different types of refrigeration systems proposed in this report. They illustrate *how* specifications are written and what they should include. They could be used as a guide for writing the job specifications to furnish and install the refrigeration equipment in any food distribution center.

Of the refrigeration specifications, those in section A contain general conditions that pertain to all situations; those in section B pertain specifically to Situation I; and those in section C pertain specifically to Situations II and III.

Specifications are also included for air conditioning, in section D; and for cold-storage doors, in section E. Section E includes the number of doors used and the individual costs. This information supplements the bills of materials included in previous sections.

These specifications *in no way* restrict or recommend the products of any individual manufacturer.

A—Typical Refrigeration Specifications—General Conditic

ARTICLE 1: CONTRACT DOCUMENTS

The contract includes the Agreement and its General Conditions, the Drawi
and the Specifications. Two or more copies of each, as required, shall be signed
both parties and one signed copy of each retained by both parties.

The intent of these documents is to include all labor, materials, appliances,
services of every kind necessary for the proper execution of the work, and the te
and conditions of payment therefor. The documents are to be considered as
and whatever is called for by any one of the documents shall be as binding a
called for by all. Any defects or inconsistencies in any document that can prev
satisfactory performance, as specified, should be brought to the attention of
Engineer prior to bidding.

ARTICLE 2: MATERIALS, APPLIANCES, EMPLOYEES

Except as otherwise noted, the Refrigeration Contractor shall provide and

for all materials, labor, tools, water, power, and other items necessary to complete the work.

All materials shall be new, and both workmanship and materials shall be of good quality.

All workmen and subcontractors shall be skilled in their trades.

ARTICLE 3: PERMITS AND REGULATIONS

Permits and licenses necessary for the prosecution of the work shall be secured and paid for by the Contractor. The Contractor shall comply with all laws and regulations bearing on the conduct of work and shall notify the Engineer if the drawings and specifications are at variance therewith.

ARTICLE 4: PROTECTION OF WORK, PROPERTY, AND PERSONS

The Contractor shall adequately protect the work, adjacent property, and the public, and shall be responsible for any damage or injury due to his act or neglect.

ARTICLE 5: ACCESS TO WORK

The Contractor shall permit and facilitate observation of the work by the Owner, his agents, and public authorities at all times.

ARTICLE 6: CHANGES IN THE WORK

The Owner may order changes in the work, the Contract Sum being adjusted accordingly. All such orders and adjustments shall be in writing. Claims by the Contractor for extra cost must be made in writing before executing the work involved.

ARTICLE 7: CORRECTION OF WORK

The Contractor shall reexecute any work that fails to conform to the requirements of the contract and that appears during the progress of the work, and shall remedy any defects due to faulty materials or workmanship which appear within a period of 1 year from the date of completion of the contract. The provisions of this Article apply to work done by subcontractors as well as to work done by direct employees of the Contractor.

ARTICLE 8: OWNER'S RIGHT TO TERMINATE THE CONTRACT

Should the Contractor neglect to prosecute the work properly or fail to perform any provision of the contract, the Owner, after seven days' written notice to the Contractor and his surety, if any, may, without prejudice to any other remedy he may have, make good the deficiencies and deduct the cost thereof from the payment then or thereafter due the Contractor; or at his option, the Owner may terminate the contract and take possession of all materials, tools, and appliances and finish the work by such means as he sees fit. If the unpaid balance of the contract price exceeds the expense of finishing the work, such excess shall be paid to the Contractor; but if such expense exceeds such unpaid balance, the Contractor shall pay the difference to the Owner.

ARTICLE 9: CONTRACTOR'S RIGHT TO TERMINATE CONTRACT

Should the work be stopped by any public authority for a period of 30 days or more through no fault of the Contractor; or should the work be stopped through act or neglect of the Owner for a period of 7 days; or should the Owner fail to pay the Contractor any payment within 10 days after it is due; then the Contractor, upon 7 days' written notice to the Owner, may stop work or terminate the contract and recover from the Owner payment for all work executed and any loss sustained and reasonable profit and damages.

ARTICLE 10: PAYMENTS

Payments shall be made as provided in the agreement. The making and acceptance of the final payment shall constitute a waiver of all claims by the Owner, other than those arising from unsettled liens or from faulty work appearing thereafter, as provided for in Article 7, and of all claims by the Contractor except any previously made and still unsettled. Payment otherwise due may be withheld on account of defective work not remedied, liens filed, damage by the Contractor to others not adjusted, or failure to make payments properly to subcontractors or for materials or labor.

ARTICLE 11: CONTRACTOR'S LIABILITY INSURANCE

The Contractor shall maintain such insurance as will protect him from claims under workmen's compensation acts and other employee benefits acts; from claims for damages because of bodily injury, including death; and from claims for damages to property which may arise both out of and during operations under this contract, whether such operations be by himself or by any subcontractor or anyone directly or indirectly employed by either of them. This insurance shall be written for not less than any limits of liability specified as part of this contract. Certificates of such insurance shall be filed with the Engineer prior to starting work on this project.

ARTICLE 12: OWNER'S LIABILITY INSURANCE

The Owner shall be responsible for and at his option may maintain such insurance as will protect him from his contingent liability to others for damages because of bodily injury, including death, which may arise from operations under this contract, and any other liability for damages which the Contractor is required to insure under any provision of this contract.

ARTICLE 13: FIRE INSURANCE WITH EXTENDED COVERAGE

The Owner shall effect and maintain fire insurance with extended coverage upon the entire structure on which the work of the contract is to be done, to 100 percent of the insurable value thereof, including: (1) items of labor and materials connected therewith, whether in or adjacent to the structure insured; (2) materials in place or to be used as part of the permanent structure, including surplus materials, shanties, protective fences, bridges, temporary structures, and miscellaneous materials and supplies incident to the work; and (3) such scaffoldings, stagings,

towers, forms, and equipment as are not owned or rented by the Contractor, the cost of which is included in the work cost

EXCLUSIONS: The insurance does not cover: (1) tools owned by mechanics; (2) any tools, equipment, scaffolding, staging, towers, and forms owned or rented by the Contractor, the cost of which is not included in the cost of the work; or (3) any temporary housing.

ARTICLE 14: LIENS

The final payment shall not be due until the Contractor has delivered to the Owner a complete release of all liens arising out of this contract, or receipts in full covering all labor and materials for which a lien could be filed, or a bond satisfactory to the Owner indemnifying him against such liens.

ARTICLE 15: SEPARATE CONTRACTS

The Owner has the right to let other contracts in connection with the work, and the Contractor shall properly cooperate with any such other contractors.

ARTICLE 16: THE ENGINEER'S STATUS

The Engineer shall be the Owner's Representative during the construction period. He has authority to stop the work if necessary to assure its proper execution. He shall certify to the Owner when payments under the contract are due and the amounts to be paid. He shall make decisions on all claims of the Owner or Contractor. All his decisions are subject to arbitration.

ARTICLE 17: ARBITRATION

Any disagreement arising out of this contract or from the breach thereof shall be submitted to arbitration, and judgment upon the award rendered may be entered in the court or the forum, State or Federal, having jurisdiction. It is mutually agreed that the decision of the arbitrators shall be a condition precedent to any right of legal action that either party may have against the other. The arbitration shall be held under the Standard Form of Arbitration Procedure of the American Institute of Architects or under the Rules of the American Arbitration Association.

ARTICLE 18: CLEANING UP

The Contractor shall keep the premises free from accumulation of waste material and rubbish and at the completion of the work shall remove from the premises all rubbish, implements, and surplus materials, and leave the building broom clean.

B—Typical Refrigeration Specifications for Situation I —Package Systems

1.01 AIR UNITS

Air units shall be of the type and size shown on the drawings. They shall be of the ceiling-hung type with heavy-duty mounting channels, with motor brackets, housing, and coils directly bolted to the channels. Housing shall b of aluminum or galvanized steel, the drain pans shall be of galvanized stee and the coils shall be of ____-inch diameter copper tubing with mechanicall bonded aluminum fins. In all rooms 40° F. or lower, coils with four fins pe inch shall be used.

Motors shall be single phase, and shall have built-in thermal overloa protection, U.L. approved.

Rooms 32° F. and below shall be equipped with a hot-gas defrosting syster as shown on the drawings, and shall be fitted as above with the followin additions.

The pan shall be steel, with a steel defrost coil electrically welded to i assembly to be hot-dipped galvanized to prevent rust and to form a solic permanent bond for good heat transfer of defrost coil to pan.

The unit shall have an accumulator designed to prevent the liquid sluggin of compressor and the trapping of oil during the defrost cycle and at r sumption of the refrigeration cycle. The accumulator shall have a built-i heat exchanger for use during the normal refrigerating cycle.

Control of the defrost cycle shall be automatic: defrosting of the coi will be started by a time clock and stopped by temperature control whe coils are completely defrosted. A single solenoid valve shall be supplied i the defrost line to regulate the flow of hot gas for defrosting.

Units shall have a fan delay.

1.02 CONDENSING UNITS

Condensing units shall be of the type and capacity shown on the drawing All units shall be of the direct-drive compressor type and shall be mounte on a raised base. All units shall be air cooled, with vertical-type coils di charging air in a horizontal plane.

The compressor shall have removable heads installed on the unit for eas removal; easily accessible valve plates; oil check valve; and dynamicall balanced rotating parts; and shall be complete with suction strainer an suction and discharge valves.

Compressors under 7½ hp. shall be lubricated by an oil slinger feedin a central oil hole in the crankshaft; those 7½ hp. and larger shall have reversing-gear-type oil pump with an oil protection switch.

Units shall be coupled to standard NEMA frame motors, as specified, vi a flexible-strap-type or a Thomas-type coupling. Final alinement shall b checked in the field.

All units shall be equipped with a pressure stabilizer to automaticall maintain a satisfactory head pressure in low ambient conditions, thus assurin proper expansion valve performance.

All units shall have suitable openings for equalizing of crankcase pressure and oil levels when units are paralleled.

Receivers shall be mounted inside each unit.

1.03 SIGHT GLASS AND DRIER

All systems are to be fitted with an angle-type drier equipped with changeable cartridges; a sight glass with moisture indicator shall be in series after the drier. Both the drier and the sight glass shall have line valves plus a suitable bypass for servicing.

1.04 THERMAL EXPANSION VALVES

Each air unit shall be fitted with a thermal expansion valve. These shall be installed in accordance with the manufacturer's recommendations.

1.05 VALVES

All valves shall be designed for R-12 or R-22 service. Valves for use with copper lines may be brass packless type, or brass seal-cap type.

1.06 PIPING

Pipe

1. Soft-temper tubing is recommended where bending is required, where tubing is to be hidden, and where flare joints are used. This tubing shall be copper, type L or K, bright annealed, dehydrated, and sealed.
2. Hard-drawn tubing shall be used for silver brazed lines where no bending is required. This tubing shall be type B copper pipe.

Joints

1. Copper tubing joints up to and including ⅝ inch may be flared or silver brazed. Silver-brazed joints should be used for all sizes larger than ⅝ inch.
2. Flare joints shall be made with flaring tools. Silfos, Easy Flow, or equivalent silver brazing wire should be used on brazed joints.

Fittings and Flanges

1. Fittings for flare joints shall be standard SAE forged brass flare type. Flare nuts shall be short-shank type.
2. Fittings for brazed joints shall be wrought copper or forged brass sweat fittings. Never use cast sweat fittings.

Hangers

1. Pipe or tubing shall be supported by split-ring adjustable-type or other suitable hangers, hung on round steel rods, or equal. Brackets or clamps may be used where lines run along walls, columns, or ceilings.
2. Valves shall be supported independently when located in copper lines smaller than 1 inch.
3. Pipe hangers shall be placed not more than 8 to 10 feet apart. If wall or ceiling brackets are used on straight lengths of pipe over 20 feet long, they shall provide for contraction and expansion. Hangers shall be placed not more than 24 inches from each change of direction, preferably on the side with the longest run.
4. Pipe hangers or brackets shall be properly isolated, where necessary, to prevent noise transmission. Never locate rigid hangers closer than 6 inches to isolated compressors.
5. Hanger rods of the following sizes, or equivalent, are recommended:

¾- to 2-inch pipe	⅜-inch rod
2½- to 3-inch pipe	½-inch rod
4- to 5-inch pipe	⅝-inch rod
6-inch pipe	¾-inch rod
8- to 10-inch pipe	⅞-inch rod
12- to 14-inch pipe	1-inch rod
14-inch pipe and larger	1⅛-inch rod

6. When pipe lines are to be insulated, the size and position of hangers shall be such as to bear on the outside of the insulation. Sleeves of No. 18 gage galvanized steel shall be placed between hangers and insulation. These shall extend at least 2 inches on each side of the hanger.

Pipe Sleeves

1. Sleeves shall be placed in floors and walls through which pipe lines pass, and extend 1 inch on each side. These may be made of pipe or of formed, galvanized steel.
2. Curbs shall be used around pipe sleeves in floors.
3. Openings are to be properly filled between the sleeve and the wall, floor, or ceiling opening.
4. Sleeves shall be furnished by the piping contractor and installed by the building contractor.

Tests

1. Refrigerant tubing and fittings shall be pressure tested with at least 50 p.s.i. pressure before charging. Pressure may be applied with the Refrigerant-22, with a mixture of Refrigerant-22 and nitrogen, or with nitrogen alone.
2. All joints shall be rechecked for leaks with full operating pressure after charging.
3. Piping must be free from leaks at test pressures. Defective material must be replaced and leaks properly repaired. Caulking or other temporary measures will not be permitted.

Construction Notes

1. The piping shall be run generally as indicated on drawings and instructions, care being taken to avoid interference with other piping, electric conduits, pneumatic tubes, etc.
2. Each part of the system of piping shall be complete in all detail, and provided with all control valves, etc., that are necessary for satisfactory operation.
3. All pipe lines shall be at least 7½ feet above floors, unless against walls or ceiling and unless requirements of refrigerant flow demand otherwise.

4. All lines shall be run plumb and straight, and parallel to walls, except that horizontal suction lines, discharge lines, and condenser to receiver lines shall be pitched in the direction of flow. Valves in these lines should be placed with stems horizontal to avoid liquid traps and damming.
5. Pockets, unnecessary traps, turns, and offsets shall be avoided. Traps or pockets, where unavoidable, shall have oil legs and drain valves.
6. Sufficient unions, flanged valves, or fittings shall be provided for disconnecting equipment, controls, etc. All piping shall be accessible for repairs.
7. Provision shall be made for contraction and expansion of three-fourths inch per 100 feet of pipe.
8. Space between pipe lines to be insulated shall be at least three times the insulation thickness for screwed fittings and four times the insulation thickness for flanged fittings. Space between pipe and adjacent surfaces shall be three-fourths these amounts.
9. All work shall be done to conform with local codes, and the necessary permits shall be obtained by the Contractor or Purchaser as provided in the contract.
10. Gages are to be installed in the suction and discharge headers and piped to a location where they can be observed by operating personnel.
11. The Refrigeration Contractor shall color code all piping with varied colored plastic bands suitable to the Engineer. All valves shall be tagged with a metal tag bearing a valve number and function.

    The Refrigeration Contractor shall furnish a piping isometric showing all main pieces of equipment and interconnecting lines bearing their respective color codes. The isometric shall be framed under glass and displayed prominently in the respective engine rooms.

1.07 EVACUATION AND CHARGING OF SYSTEM

All R–12 and R–22 systems shall be evacuated and charged by the triple evacuation method. A vacuum pump capable of pulling the system down to the 50- to 100-micron range shall be used. The pump shall be equipped with an electronic gage.

During the first evacuation, the system shall be pulled down to the 100-micron range and held there for 3 to 4 hours, after which it is charged with dry nitrogen or a mixture of nitrogen and refrigerant. In small systems, a small quantity of refrigerant can be used for charging.

For the second evacuation, repeat the procedures followed in completing the first evacuation and charging.

For the third evacuation, the system is again pulled down to the 100-micron range and held there for 3 to 4 hours. This time, however, the amount and type of refrigerant to be used in the operation of the system is used for the charging.

C—Typical Refrigeration Specifications for Situations II and III—Central Systems

1.01 SCOPE

Under this contract shall be provided a complete central system, ammonia, R-717, pump-feed liquid-recirculation system as shown on the drawings, No. _____ through No. _____, dated __________. Room conditions, as indicated, must be obtainable with the loading, as specified. Each room shall be provided with air units, as specified, with full provision made for proper servicing of same. The engine room will be equipped as shown on the drawings. Any omissions or errors in the documents are to be brought to the attention of the Engineer prior to bidding.

1.02 SUBSTITUTIONS

All bids are to be based on the plans and specifications, and shall be so bid. Any substitutions must be listed as an addition or deduction to the base bid, and must be accompanied by sufficient information, capacities, dimensions, etc., so that they can be evaluated by the Engineer.

1.03 AIR UNITS

All air units are to be of the ceiling-hung type, with hot-dipped galvanized coils constructed of steel fins bonded to steel tubes. Coils shall not have more than four fins per inch, except in rooms of 50° F. or higher. Coils shall be circuited for liquid recirculation, and shall state the required g.p.m. for proper operation at rated capacities. Housing shall be aluminum or galvanized steel. Units shall meet ratings and specifications as shown on the drawings.

Units furnished for cutting rooms and other work areas shall have filters of the throwaway glass-fiber type preceding the coils.

Units furnished for 32° F. rooms and below shall be equipped to use hot gas for defrosting the cooling coils and drain pans. Units shall have single-outlet drain pans fitted with a hot-gas coil. The hot-gas inlet to the air unit shall be mechanically secured to the drain-pan outlet. A check valve shall be installed between the outlet of the hot-gas coil in the drain pan and the hot-gas inlet to the cooling coil.

The air units in the fresh-meat holding rooms shall have a stainless-steel housing and drain pan, and be equipped for hot-gas defrost.

1.04 AUTOMATIC VALVES

Automatic valves shall be as specified on the drawings. Valves shall be of all-steel construction. Solenoid valves are to be equipped with pilot lights to indicate when the valve is energized.

Suction solenoid valves in rooms below 0° F. are to be pilot-operated, spring-loaded valves and *cannot* incorporate a relief valve or regulating valve operation.

Relief valves on the suction side of a hot-gas defrost system shall have

chrome seats, and shall be equipped with a cock and valve for ease in adjusting settings. Relief valves and regulators can be incorporated into the suction solenoid valves when the room temperature is above 0° F.

1.05 DEFROST TIMERS

Defrost timers shall be of the 24-hour type, with a minimum of six defrosting periods available. They shall incorporate a pumpout feature, up to a 60-minute defrost cycle, and a fan delay feature on startup. They shall be installed in a waterproof casing suitable for installation in or out of the cold room.

1.06 PUMP ACCUMULATORS AND INTERCOOLERS

Pump accumulators and intercoolers shall be constructed to the dimensions and contain the connections as shown on the drawings. Shells shall be constructed and inspected in accordance with the American Society of Mechanical Engineers (ASME) Unfired Pressure Vessel Code, with design working pressure of 150 p.s.i. Exterior surfaces shall have one coat of rust-resistant paint, and all connections shall be plugged after fabrication.

1.07 LIQUID AMMONIA PUMPS

Pumps shall be of the capacity and type shown on the drawings. Pumps shall be positive-displacement gear pumps of the rotary type and shall be V-belt driven, with drives as per the drawings. Pumps shall be equipped with a double mechanical seal pressurized by refrigeration oil from an oil reservoir. Sight glass with frost shields shall be installed on the oil reservoir. An oil-filling valve shall be furnished so that oil can be added without stopping the pump.

1.08 COMPRESSORS

Compressors shall be of the capacity and type shown on the drawings, and shall be of the multicylinder direct-drive type. Housing shall be of close-grained iron casting; shall be complete with hand-hole plates, oil sight glass, suction and discharge valves, suction strainer, and internal relief valves; and shall have removable cast alloy iron sleeves fitted into the cylinders.

Pistons are to be of the cast-iron plug type or double-trunk slipper type with chrome-plated compression rings, and shall be fitted with aluminum alloy, permanent-mold-cast connecting rods with integral bearing in crank end. The lubrication system shall be of the forced-feed type with a reversible-type pump; and shall be fitted with an oil-pressure switch, of the differential type, with hand reset and integral time delay relay.

A spring-loaded, ball-check-type relief valve, set for 300 p.s.i., shall be installed between discharge and suction passages.

Each discharge connection shall be fitted for a discharge-gas thermometer. Each compressor shall have a discharge-line oil separator, complete with its own float, which shall return oil to the oil receiver system.

A crankcase float shall be installed inside the compressor to admit oil, as required, from the oil receiver

1.09 OIL RECEIVER

An oil receiver shall be supplied as shown on the drawings, and shall be equipped with gage glass or bulls-eye for level indication The receiver shall be equipped with a thermostatically controlled 300-watt heater to maintain a temperature of 90° F.

1.10 COMPRESSOR JACKET COOLING

Compressor jacket coolers of the Refrigerant–11 type, as shown on the drawings, shall be installed as shown The water coolant for the R–11 shall be controlled by a temperature-actuated water valve controlled by a relay and thermostats installed in the gas line at the compressor jackets Waste water shall be piped to the basins of the evaporative condensers

1.11 EVAPORATIVE CONDENSERS

Evaporative condensers shall be of the type and capacities as shown on the drawings They shall be constructed either of galvanized 11-gage steel sheet, followed by welding the seams and painting them with rust-resistant aluminum paint; or of 11-gage steel sheets which shall be hot-dipped galvanized after fabrication.

The fan drive shall be V-belt, and shall have a protective guard with adjustable belt tension.

The condensing coil shall be of the staggered-tube design with ___-inch full-weight steel pipe. The coil shall be of completely welded construction and shall be hot-dipped galvanized after fabrication and tested under water to 300 p.s.i.g. of air.

Eliminators shall be galvanized steel, minimum of ___ gage Spray nozzles shall be removable, bronze, nonclogging centrifugal, two-piece design, installed in full-weight galvanized pipe system. A valved bleed connection shall be provided for constant bleedoff.

A spray water pump of the centrifugal type shall be furnished with screens at the suction outlet of each pan.

Expanded metal air-inlet screens shall cover all air inlets

Discharge air dampers are to be installed on all units installed inside, and are to be controlled by head pressure.

Fans are to be cycled from end switches on these dampers

Water pumps shall run at all times except when head conditions cannot be maintained during extremely cold weather.

1.12 HIGH-PRESSURE AMMONIA RECEIVERS

Receivers shall be constructed in accordance with ASME Code and inspection procedures, and shall be tested in accordance with American Standards Association (ASA) B-9 Code. Design working pressure shall be 300 p.s.i. for shells 24 inches in diameter or less, and 250 p.s.i. for those 30 inches in

diameter or larger. All receivers shall be sized and fitted in accordance with the drawings; complete with stands, liquid gages and valves, charge and drain valves, and inlet and outlet valves; and equipped with dual relief assemblies set at 250 p.s.i. with three-way valve.

.13 AMMONIA PIPING

Pipe

1. Ammonia lines shall be constructed of seamless or lap-welded steel pipe. Butt-welded pipe may be used for sizes 1½ inches and smaller.
2. All pipe 1 inch and smaller should be extra heavy. Suction and discharge lines larger than 1 inch may be full weight.
3. Liquid lines 1½ inches and smaller should be extra heavy. Sizes larger than 1½ inches may be full weight.

Joints

1. Joints between lengths of pipe or between pipe and fittings shall be threaded for pipe sizes 1½ inches and smaller. Sizes larger than 1½ inches shall be welded. Sizes smaller than 1½ inches may be welded when preferable.
2. Threads shall be of standard taper. They shall be cut clean and free of burrs. Burrs formed inside pipe from cutting shall be removed by reaming. Joints shall be made with litharge and glycerine or other suitable joint compound.
3. Welded joints shall be made by experienced welders. Welding surfaces shall be cleaned and properly spaced before welding. Parts to be welded shall be in line at all points. A weld 2½ times the wall thickness of the pipe is recommended.

Fittings and Flanges

1. Fittings and flanges shall be ammonia-type.
2. The type and finish of fittings and flanges shall correspond to the selection of pipe and type of joint.
3. Flanged fittings shall be used for pipe sizes larger than 2 inches, except that screw-and-flange type may be used up to 3 inches inclusive.
4. Flanges and flange faces of fittings shall be tongue-and-groove type. Sizes 1 inch and smaller may be oval (two-bolt) type. Sizes larger than 1 inch, up to 4 inches inclusive, shall be square (4-bolt) type. Sizes larger than 4 inches shall be round.
5. Steel welding neck flanges and welding elbows shall be used for welded pipe lines.
6. Pipe bends may be used in lieu of welding elbows where space permits. Socket weld fittings are recommended for sizes 1 inch and smaller.
7. Unions shall be flanged type.
8. Bushings may be used for reductions of two or more pipe sizes. Reducing fittings should be used where the reduction is only one pipe size. Bushings 2 inches and smaller shall be steel. Sizes larger than 2 inches may be malleable or air-furnace iron.

Valves

1. Valves shall be air-furnace iron or steel-body ammonia globe or angle type. Gate valves may not be used for ammonia service.
2. Valves 2 inches and smaller may be screwed type. Flanged valves should be used in sizes larger than 2 inches.
3. Valves 1 inch and smaller may have screwed bonnets. Sizes larger than 1 inch shall have bolted bonnets.
4. Steel angle valves one-half inch and smaller used for drain, purge, or gage lines need not have back seats. All other valves shall be back seated for repacking in service.
5. Check valves may be lift or swing-check type. Sizes larger than 1 inch shall have manual lifting stems.

Gaskets

Gaskets shall be 1/16-inch thick asbestos fiber composition or soft lead, and shall fit accurately into grooves of fittings.

Hangers

1. Pipe shall be supported by split-ring adjustable type or other suitable hangers hung on round steel rods, or their equal. Brackets or clamps may be used where lines run along walls, columns, or ceilings.
2. Pipe hangers shall be placed not more than 8 to 10 feet apart. If wall brackets are used on straight lengths of pipe over 20 feet long, they shall provide for contraction and expansion. Hangers shall be placed not more than 24 inches from each change of direction—preferably on the side with the longest run.
3. Pipe hangers shall be properly isolated, where necessary, to prevent noise transmission.
4. Pipe-hanger rods of the following sizes, or equivalent, are recommended:

¾- to 2-inch pipe	⅜-inch rod
2½- to 3-inch pipe	½-inch rod
4- to 5-inch pipe	⅝-inch rod
6-inch pipe	¾-inch rod
8- to 10-inch pipe	⅞-inch rod
12- to 14-inch pipe	1-inch rod
14-inch pipe and larger	1⅛-inch rod

5. When pipe lines are to be insulated, the size and position of hangers shall be such as to bear on the outside of the insulation. Sleeves of No. 18 gage galvanized steel shall be placed between hangers and insulation. These shall extend at least 2 inches on each side of the hanger.

Hanger inserts shall be installed in concrete ceilings of new construction. These shall be supplied by the piping contractor for installation by the building contractor.

e Sleeves

Sleeves shall be placed in floors and walls through which pipe lines pass, and extend 1 inch on each side. These may be made of pipe or formed galvanized steel.
Curbs shall be used around pipe sleeves in floors.
Openings are to be properly filled between sleeve and the wall, floor, or ceiling opening.
For new construction, sleeves shall be furnished by the piping contractor and installed by the building contractor.

ts

Piping, after installation, shall be tested with 300 p.s.i. air pressure or nitrogen on high-pressure side, and 150 p.s.i. on low-pressure side. Piping must be free from leaks at these pressures.
Defective material must be replaced and leaks properly repaired. Caulking or other temporary measures will not be permitted.
Valves and cast fittings shall be factory tested with 300 p.s.i. air pressure under water.

struction Notes

The piping shall be run generally as indicated on drawings and instructions, care being taken to avoid interference with other piping, electric conduits, pneumatic tubes, etc.
Each part of the system of piping shall be complete in all detail and provided with all control valves, etc., necessary for satisfactory operation.
All pipe lines shall be at least 7½ feet above floors, unless against walls or ceiling.
All lines shall be run plumb and straight, and parallel to walls, except that horizontal liquid lines between condensers and receivers and all low-pressure liquid lines shall pitch one-fourth inch per foot in the direction of flow.
Pockets, unnecessary traps, turns, and offsets shall be avoided. Traps or pockets, where unavoidable, shall have oil legs and drain valves.
Sufficient unions, flanged valves, or fittings shall be provided for disconnecting equipment, controls, etc. All piping shall be accessible for repairs.
Provision shall be made for contraction and expansion of three-fourths inch per 100 feet of pipe.
The distance between pipe lines to be insulated shall provide ample working space for handling insulation. It is recommended that the space be at least three times the insulation thickness for screwed fittings and four times the insulation thickness for flanged fittings. Space between pipe and adjacent surfaces shall be three-fourths these amounts.

9. All work shall be done to conform with local codes, and the necessary permits shall be obtained by the Contractor or Purchaser as provided in the contract.
10. After the Painting Contractor and Insulation Contractor have completed their contracts, the Refrigeration Contractor shall color-code all piping with varied colored plastic bands suitable to the Engineer. All valves shall be tagged with a metal tag bearing a valve number and function.
11. The Refrigeration Contractor shall furnish a piping isometric showing all main pieces of equipment and interconnecting lines bearing their respective color codes. This isometric shall be framed under glass and displayed prominently in the respective engine rooms.

1.14 AUTOMATIC PURGER

A continuous automatic purger of the inverted-bucket type shall be furnished for each booster ammonia-refrigeration system. The purger shall be complete with automatic expansion valve, hand valves, and water container for vented gases.

1.15 AMMONIA METERS

Ammonia meters shall be supplied on all main liquid lines of each building and in the high-stage and low-stage liquid lines to each of the firms using refrigeration. The meters shall be located where dial indicators can be read or supplied with a remote indicator.

Meters shall be constructed of cast iron or stainless steel housings with hard-carbon bearings. Units shall be flanged with screwed or welded connections. Meters shall be properly sized and accurate to ± 0.5 percent.

D—Typical Air-Conditioning Specifications for Situations II and III

2.01 FAN COIL UNITS

The coil shall be fabricated from ____-inch OD copper tubes, with a ____-inch-thick wall, and from ____-inch-thick aluminum flat fins, arranged with ____ fins/inch. The coil shall be ____ rows deep and ____ rows high; and shall have two feeds per coil and a ⅛-inch Iron Pipe Tap (IPT) air vent. The coil shall be tested for a maximum operating pressure of 250 p.s.i. The fan deck will be constructed of ____-gage galvanized steel, assembly spot welded, and with one removable inlet ring per fan housing.

Fan wheels shall be of the forward-curve blade design, and constructed of aluminum blades and center disk with steel hub.

Fan motors shall be 115/1/60 volt nominal, permanent-split capacitor motors, with oil tubes and inherent overload protection.

The drain pan shall be fabricated from ____-gage galvanized steel, and shall extend or have an auxiliary pan under the control valves so that all condensate produced within the enclosure shall be drained. Pan should be insulated with a minimum of ½-inch polyester insulation. Enclosure shall be fabricated of ____-gage furniture steel, bonderized and spray painted, with a baked-enamel finish. Access doors to valves and controls shall be provided.

The discharge grill shall be of the double-deflection type.

The enclosure shall be insulated or designed to eliminate all sweating. A three-way water control valve shall be installed for temperature control and shall be controlled by a wall or self-contained thermostat as per the owner's request.

2.02 HEAT EXCHANGER

The shell shall be fabricated of seamless or resistance-welded pipe up to 24 inches in diameter, and of formed steel with fusion-welded longitudinal seam above 24 inches in diameter.

Tube sheets shall be steel, fusion-welded to shells at least 1¼-inch thick, and shall have tube holes reamed and recessed.

Tubes shall be 1¼-inch OD No. 13 Birmingham Wire Gage Electric Resistance welded steel of American Society for Testing and Materials Specification A-214.

Water heads are to be of cast iron with machined gasket surfaces. Design maximum working pressure:

24 inches and under	300 p.s.i.
Over 24 inches	250 p.s.i.
Water heads	150 p.s.i.

Shell and water passes shall be tested in accordance with ASA B–9 code. Shell and tube subassemblies shall be submerged under water, after fabrication, and tested with 250 p.s.i. air pressure.

Shells are to be constructed and inspected in accordance with ASME Unfired Pressure Vessel Code.

A level-control switch for summer operation shall be provided to open and close the liquid-inlet solenoid. This switch shall be a float switch suitable for operation at 250 p.s.i.

A back-pressure regulator and stop valves shall be used to control the water temperature. A safety thermostat set for 35° F. shall be provided in the leaving water system. Interlocks through the water pumps shall supply the voltages to these valves as a further safety precaution.

A manually operated valve shall be installed to switch the heat exchanger from the low side to the high side for winter operation. A manual switchover shall be provided on the control circuit.

Head pressure shall be set to supply the temperature of water required, depending on ambient conditions. A step controller shall shut down evaporative condenser fans and follow by sequencing the pumps on the evaporative condenser water systems. In extreme weather the heat exchanger shall furnish the only condensing action in the circuit except for that due gravity losses in the evaporative condensers, which is very small.

2.03 WATER PUMPS

Pumps of the type and capacities shown on the drawings shall be furnish The case shall be cast iron, vertically split, and shall have flanged suct and discharge connections. The impeller shall be bronze, dynamic balanced, of the enclosed type, and locked to the pump shaft. The shaft s shall be of the rotary mechanical type suitable for water temperatures up 250° F. The motor shall be sized so that the maximum brake horsepo output required by the centrifugal pump is less than that allowed by motor manufacturer's guarantee service factor. The motor shall be c nected to the pump shaft with a flexible self-alining coupling. The pump sl have sleeve bearings and force-feed lubrication. The bearing-bracket assem shall be removable without disturbing the piping or the motor. In the wet area the pump shaft shall be protected by a nonferrous sleeve. The base sl be constructed of structural angles with integral feet, and shall be open the top to reduce noise. The pump shall be suitable for 175 p.s.i. work pressure.

2.04 VALVES AND PIPING

Piping shall be of Type L copper, and installed in accordance with sp fications for piping as in sections B and C.

Valves shall be of galvanized or nonferrous construction, and suitable working pressures of 150 p.s.i.

E—Typical Cold-Storage Door Specifications and Installation Schedule

DOOR No. 1: Manual Single Horizontal-Slide Cooler Door, complete with _ -gage galvanized steel door, unpainted; 4-inch polyureth foamed-in-place insulation; neoprene seals, galvanized tr mounted on painted wood header, painted wood casings, trol trucks, stay rollers, galvanized operating hardware, and lock device with inside safety release.
5′0″ × 8′0″
Unit Price Installed .... $925.00 e
No. Required: Bldg. No. 1 = 18; Bldg. No. 2 = 11; Bldg. No. 3 = 15; Bldg. No. 4 = 4.
Total = 48.

DOOR No. 2: Identical to Door No. 1, but with provision for meat rail.
Unit Price Installed .... $1,025.00 e
No. Required: Bldg. No. 2 = 12

DOOR No. 3: Identical to Door No. 1, but with heating cable for 110/1/60.
Unit Price Installed $975.00 each
No. Required: Bldg. No. 3 1

DOOR No. 4: Identical to Door No. 1, but with 6 inches of polyurethane insulation and with heating cable for 110/1/60.
Unit Price Installed $1,095.00
No. Required: Bldg. No. 2 5; Bldg. No. 3 6; Bldg. No. 4 - 2.
Total 13

DOOR No. 5: Identical to Door No. 4, but 4'0" × 7'0"
Unit Price Installed $975.00
No. Required: Bldg. No. 3 2

DOOR No. 6: Identical to Door No. 1, except 2 inches of polyurethane insulation.
Unit Price Installed $700.00
No. Required: Bldg. No. 3 = 4

DOOR No. 7: Identical to Door No. 6, except 4'0" × 7'0"
Unit Price Installed $625.00
No. Required: Bldg. No. 3 = 2

DOOR No. 8: Wooden cold-storage door, right-hand swing, complete with 4-inch insulation, galvanized hardware, and wood frame, neoprene seals, and locking device. 3'0" × 6'0".
Unit Price Installed $475.00
No. Required: Bldg. No. 1 = 2

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